HINTS ON
STEAM-ENGINE DESIGN
AND CONSTRUCTION.

*WITH PRACTICAL SUGGESTIONS FOR THE GUIDANCE
OF JUNIOR ENGINEERS AND STUDENTS.*

BY

CHARLES HURST,
AUTHOR OF "VALVES AND VALVE-GEARING," ETC.

WITH DIAGRAMS AND ILLUSTRATIONS.

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TABLE OF CONTENTS.

HINTS ON STEAM ENGINE DESIGN AND CONSTRUCTION.

CHAPTER I.

STEAM PIPES.

Introduction.—In engineering, experience is the knowledge of the behaviour of certain contrivances in actual working, and a just estimation of the difference between the result anticipated by theory and that realised in practice. It is knowledge gained by the observation and comparison of many conditions, and is perhaps to be most rapidly attained by a study of failures. To a genius it may not be essential, but an ordinary man gets his living with it.

The writer has often thought what a valuable work might be written by any successful engineer, if he had only the courage to give us a record of all his mistakes, his erroneous notions, and his ignorance. We should see how he started with very limited knowledge, how he groped in the dark, yet was continually gaining wisdom, and applying it to the work he had in hand. We could follow the working of his mind, and trace the considerations which guided him in carrying out his successful undertakings. The work would be as interesting to engineers as the diary of Mr. Samuel Pepys is to the student of literature and history. But perhaps it is too much to expect such a book. If an engineer kept such a diary he would probably leave instructions in his will that it should be burnt; or, bequeathing it to his son, it would be looked upon very much in the same light as the life of Dr. Johnson was regarded by the son of that illustrious biographer.

But though it is not to be expected that the engineering

profession will ever be enriched by such a work, it may yet be possible to write a book dealing with a few everyday questions in an engineer's business relative to matters which, though trivial and obvious when spoken of, have been, nevertheless, too often ignored. That such a work would be of some use to young engineers cannot reasonably be doubted, and may also be of aid to those who have passed these earlier stages.

As the writer's experience has been mostly concerned with steam engines, it is that branch of engineering which will be chiefly dealt with; yet, although treating of only one branch, it may be found useful and instructive to some who are about to take up the study of other classes of work, since many of the principles herein considered are common to all kinds of machines and structures. The discussions, too, of various designs, and the observation of faulty arrangements, are well calculated to induce the young engineer to trace a reason for the particular shape and dimension of every piece he designs, and gradually lead him to ask himself instinctively what objections and advantages lie in the alternatives for any given contrivance he may be engaged upon; so that he falls in the habit of mentally erecting and dismantling the work he has in hand, and becomes, in spirit at least, the engineer in charge, able to appreciate all the joys and sorrows attending such a position.

Perhaps the most valuable quality a young engineer can possess is strong common sense. After that he should be endowed with imagination. Common sense will guide him to the best methods of attaining his ideal. It will decide the proportion and form of each detail, and teach him to reject what is poor and weak in form and unpleasing in appearance. His imagination enables him rapidly to conceive contrivances, and invent methods for overcoming difficulties. He sees at a glance the general plan of the work he has in hand, and this will enable him to have the completed machine in his mind's eye and foresee its behaviour in action. If he combines with these qualities a fair proportion of industry, it will go hard with him indeed if he does not meet with his just reward, not the least part of which is the reflection that he has produced a machine or structure that not only does all it was intended to perform, but is, at the same time, presentable in appearance.

I shall commence with some remarks on the junction-valve on the boiler, then pass on to the steam pipes and the stop-valve, and so follow the course of the steam until it passes over the hotwell away to the reservoir. I then purpose going round the engine, dealing first with the motion work, the large pieces, and gradually pass on to the smallest details.

In designing machinery the engineer should, above all things, endeavour to avoid breakdowns; after that he should arrange the parts, so that the inevitable profane language on the part of the attendant engineer may be reduced to a minimum; and lastly, he may devote his energies to giving the parts a nice proportion and a good finish.

Steam Pipes.—Starting then with steam pipes, it may be remarked that any breakdown here would be most serious, endangering life and property, so that it is the duty of every engineer to devote great care to the design of the main range.

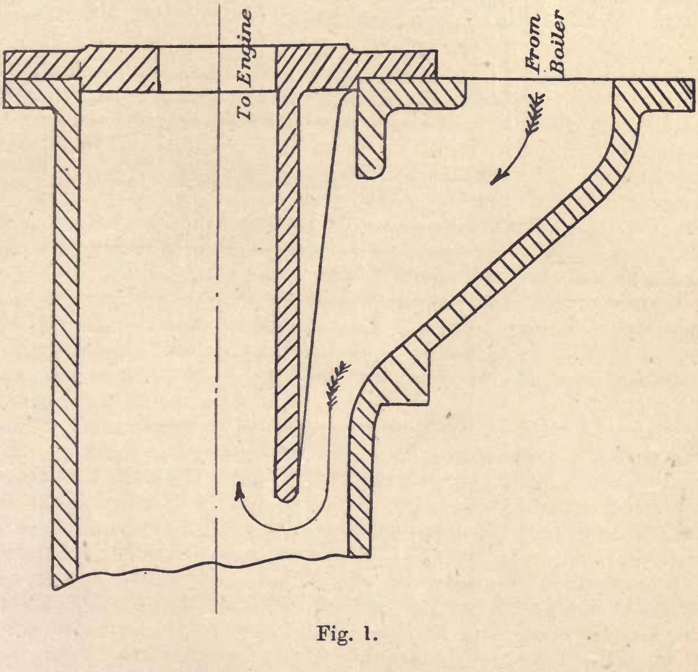


Fig. 1.

Simple as a line of piping appears, there are yet many points to observe, in order that it shall be really satisfactory. The pipes should have a fall in the direction of the current, and at the lowest point a drain pipe and steam trap should be fitted. At this point also a separator should be fitted if the length of piping from the boiler to the engine exceeds 100 feet. This separator should be as near the stop-valve as possible, and so

constructed as to change the direction of the steam completely, as shown by the figure (Fig. 1). If the above conditions are observed, there will be no danger from water hammer. Any part where water may lodge should be effectively drained ; but it will generally be found that a judicious placing of the separator will serve for the whole line of pipes.

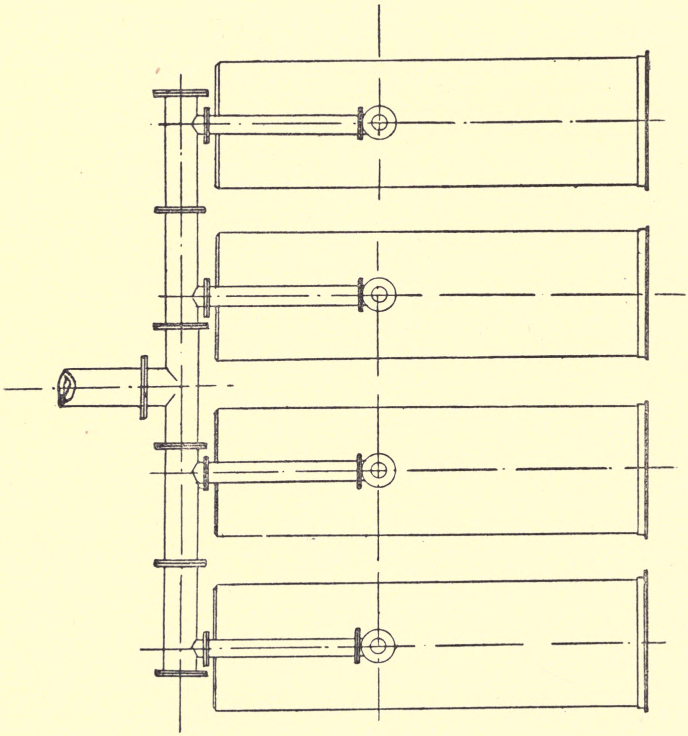


Fig. 2.

The commonest trouble with steam pipes, however, is leakage at the joints, caused in most instances by too great rigidity in the arrangement. Cast-iron contracts or expands ·0000739 in. per foot for each degree Fah. ; so that the creep in a range 30 yards long carrying steam at 100 lbs. can soon be reckoned, and will be found to work out to something like 2 inches.

Means must be provided to allow of this expansion and contraction without straining or opening any of the joints.

In Fig. 2 is shown a pipe arrangement for four Lancashire boilers actually in existence, and supplying steam to a 1200 I.H.P. mill engine. These pipes, as might be expected, have always given trouble, and it is impossible to keep all the joints tight. In this arrangement the pipes leading from the

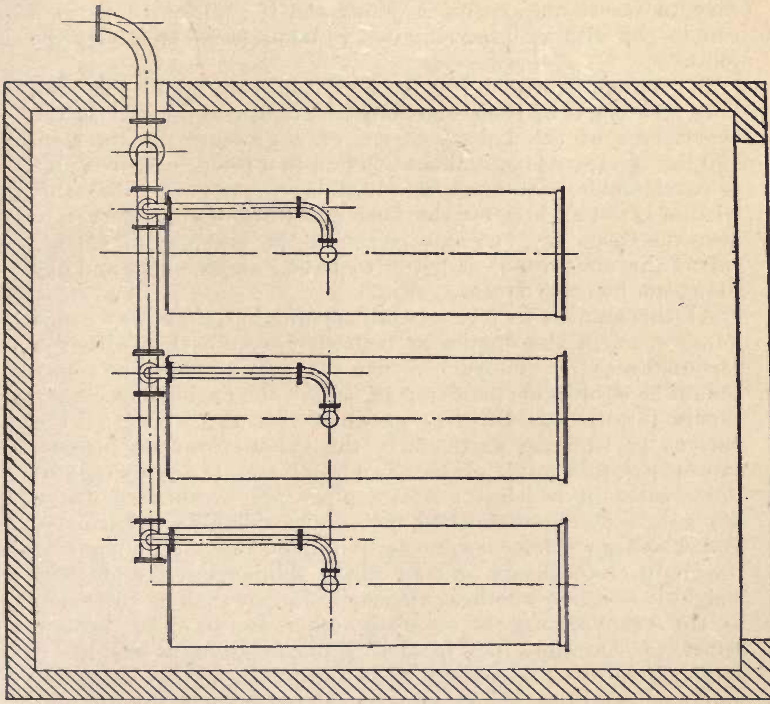


Fig. 3.

junction-valve are carried straight to the main range; the engine supply is taken from the central point and carried straight to the engine-house without bends, so that a severe stress on the flanges takes place which tends to open the joints. The whole arrangement is too rigid, and "blowing joints" are the result.

A much better plan is shown in the next figure, where three

boilers are shown working into one main range. Expansion or contraction either way is in this case not injurious, because the movement is taken up by one flange of the joint twisting or rubbing on the other, the clearance in the holes of the pipe flanges allowing of this. It will also be noted that bends, which may be reasonably expected to allow of some movement before the joints gape, are very frequent. In cases where it is impossible to obtain an elastic arrangement, it becomes necessary to insert an expansion joint, and if sufficient expansion can be got with a copper ring it is to be preferred to a telescope joint.

For pressures above 160 lbs. steel pipes are by far the best, both on account of their elasticity and lightness, as well as the smoothness of the bore. Steam of a pressure greater than 160 lbs. is very dense, and the friction in a rough cast-iron pipe is considerable. It must be remembered, however, that this friction is not all loss, for the heat generated thereby serves to keep the steam dry, but as this is not the best way of getting rid of the moisture it is better to have a smooth bore and dry the steam by other means.

At this point it may be as well to point out that if any pipe joints come in the engine or boiler-house walls they will be a certain cause of annoyance. In the boiler-house the pipes should be clear of all the dampers, and in the engine-house they should interfere as little as possible with the lifting of the various parts, more particularly the cylinder covers, pistons, piston-rods, and crank shaft. The latter remarks apply only to those cases in which the steam pipes are overhead. These are points easily overlooked, yet of considerable importance, and which go a long way towards influencing the opinions of the engineer in charge—a man whose opinion is often of great weight in deciding whether extensions of plant shall be entrusted to the original makers or elsewhere. It may be further remarked that the pipes in the engine foundations should be arranged so as to be accessible at the joints and to clear the holding-down bolts, it not being considered first-class practice to pass a bolt through a pipe; neither is it a satisfactory plan to have pipes passing through the arms of the flywheel. As to the size of the main range it is generally conceded that it should be of ample area, the maximum speed of the steam not exceeding 50 feet per second or thereabouts. In case of priming at the boilers a range of small diameter allows the steam to carry moisture over to the cylinder, which at a lower velocity would not have been held in suspension. Large pipes, of course, carry the disadvantage of increased condensing surface, so that the

heat insulation of the main steam pipes must be well carried out if economy is to be effected. A copious main also forms a reservoir from which the cylinder can draw at each stroke, and if the motion of the steam could be observed it would be seen to issue at an almost uniform rate from the boiler; whereas if the reservoir space were very small it would have a pulsating motion, and proceed in a series of rushes, with alternate pauses, to the engine. Finally, all long ranges of pipes ought to be supported by slings or brackets, so as to relieve the bolts at the joints from carrying the weight, and a clear head-room of not less than 6 feet 6 inches should be given beneath the pipes above the floor level.

The main object to be kept in view is to get the steam to the cylinder as dry as possible, wet steam being very objectionable and wasteful. It is for this reason that superheating, when properly carried out, has always been found to effect a saving, and, indeed, seems to be one of the few directions in which increased efficiency in the steam engine is to be sought.

Superheating. — In the Schmidt system some excellent results have been obtained which are worthy the attention of all interested in steam engineering. The superheating apparatus varies in its details and arrangements to suit circumstances. In the case of a plant initially designed for the system, the superheating is effected by means of the boiler furnace gases, but when applied to an existing plant the superheater is separately fired. In compound engines the system also provides for an intermediate superheating between the high- and low-pressure cylinders by passing a portion of the highly superheated steam through the receiver, the amount being varied automatically by a valve, so that as the pressure in the receiver rises the amount of highly superheated steam passing through the receiver is diminished, and *vice versa;* and it would appear from the results of tests that a high degree of initial superheating and an intermediate superheating is necessary to prevent liquefaction. Thus, in the trials of an engine at Middlepolder, near Amsterdam, an initial superheating of 214° C. and an intermediate superheating of 72° C. did not prevent a small amount of liquefaction in the low-pressure cylinder. The results of this trial are worthy of note, and a diagram showing the difference between working with saturated and superheated steam is appended (Fig. 4). The figures in full lines are those obtained when working with saturated steam at 140 lbs. by gauge; the dotted lines the result of superheating to the amount stated above. The saturation curves are also shown and indicate the condition of the steam at any point in the stroke.

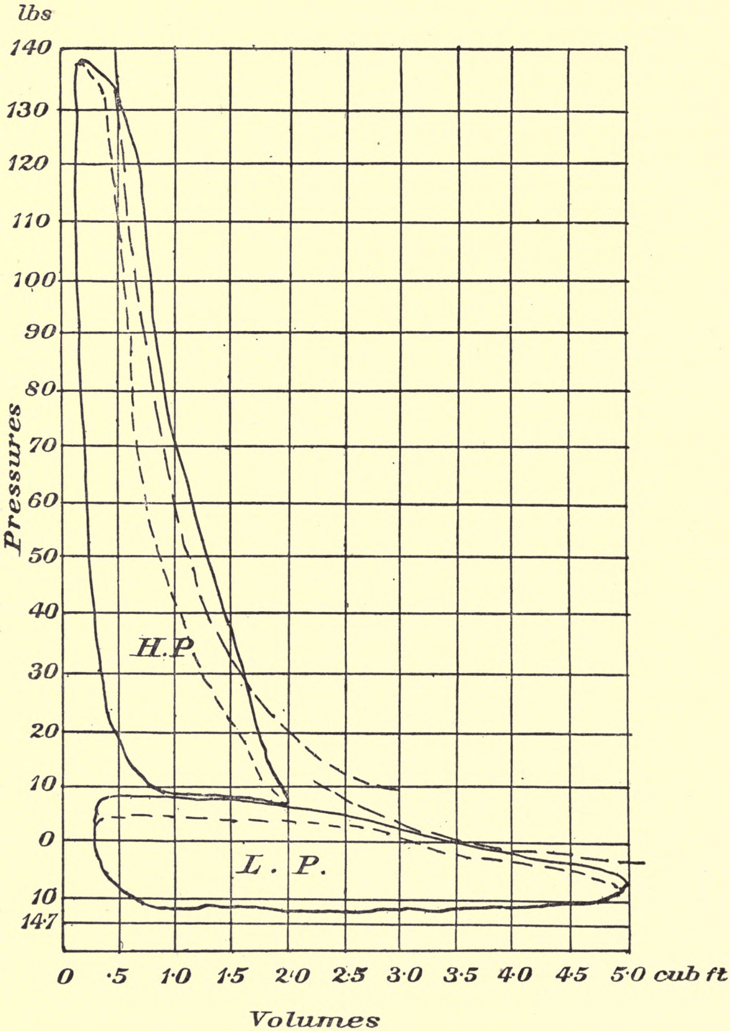

lbs
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
10
14·7
Pressures
H.P.
L. P.
0 ·5 1·0 1·5 2·0 2·5 3·0 3·5 4·0 4·5 5·0 cub ft
Volumes

Fig. 4.

In order that the full conditions may be understood the following tabulated results were observed in each trial :—

	Superheating.	Saturated.
Diameter of high-pressure cylinder,	16·14 ins.	
Diameter of low-pressure cylinder,	25·59 ins.	
Stroke of high- and low-pressure cylinders,	15·75 ins.	
Pressure of steam in boiler by gauge, lbs. per square inch,	140	140
Corresponding temperature of saturation,	182° C.	
Temperature of steam on leaving superheater,	396° C.	...
Temperature of steam in high-pressure valve-chest,	292° C.	...
Pressure of steam in receiver, lbs. per square inch,	7½	6½
Corresponding temperature of saturation,	111° C.	
Temperature of receiver steam after superheating,	183° C.	...
Vacuum,	27 ins.	27·5 ins.
Revolutions per minute,	124·2	114·6
I.H.P., high-pressure cylinder,	115·5	65·6
I.H.P., low-pressure cylinder,	68·7	59·2
Total I.H.P.,	184·2	124·8
Feed water per hour in lbs.,	1920	2143
Feed water per I.H.P. in lbs.,	10·4	17·2
Coal per hour in lbs.,	242	262
Coal per I.H.P. hour in lbs.,	1·31	2·10
Steam per lb. of coal in lbs.,	7·93	8·18
Original temperature of feed water,	20° C.	18° C.
Temperature of feed after passing economiser,	76° C.	75° C.
Temperature of chimney gases,	175° C.	185° C.

The coal consumption is the true basis for estimating the saving effected by superheating, because the boiler was fired with the same coal and under the same conditions in each trial. The consumption, then, in the saturated trial was 2·10 lbs. per I.H.P. per hour; in the superheated trial, 1·31 lbs. This, it must be admitted, is a remarkable result and thoroughly demonstrates the advantages of superheating. The report of the trials does not state whether the engine was specially designed for using superheated steam, but, from the great reduction in temperature between the boiler and the valve-chest of the high-pressure cylinder, and at other points, it would appear that no special arrangement for heat insulation had been made. This matter, however, is important, and when superheating becomes

general, as it has every prospect of doing, will demand careful attention. There can be no doubt that this trial does not by any means exaggerate the full amount of advantage to be got from the system. The report goes on to say that the lubrication was in every respect satisfactory; no abrasion of the valve faces or of the cylinder being noticed, and no apparent inconvenience was experienced from the high temperature of the steam. In the words of Professor Ewing, who made the tests, the results obtained with high superheating in large engines excel anything otherwise obtained, and small engines are enabled by its aid to reach results equal to those obtained only in the best large engines using saturated steam.

CHAPTER II.

VALVES.

Stop-Valves.—The main stop-valve of the steam engine has many forms, but the ordinary screw down valve is the most common. For small sizes and pressures not exceeding 100 lbs.

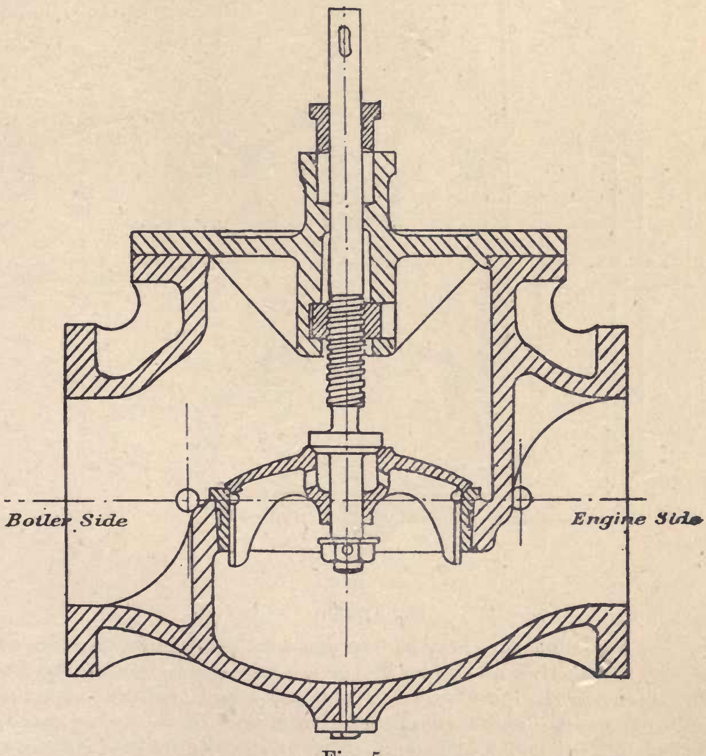


Fig. 5.

this form is quite satisfactory, but when dealing with steam of, say, 180 lbs. and upwards, and pipes anything above 8 inches

diameter, considerable skill is necessary in order to design a satisfactory valve. The chief objection to the ordinary screw down valve for high pressures is the difficulty of opening it against the steam pressure. The second objection is the tortuous passage through the valve.

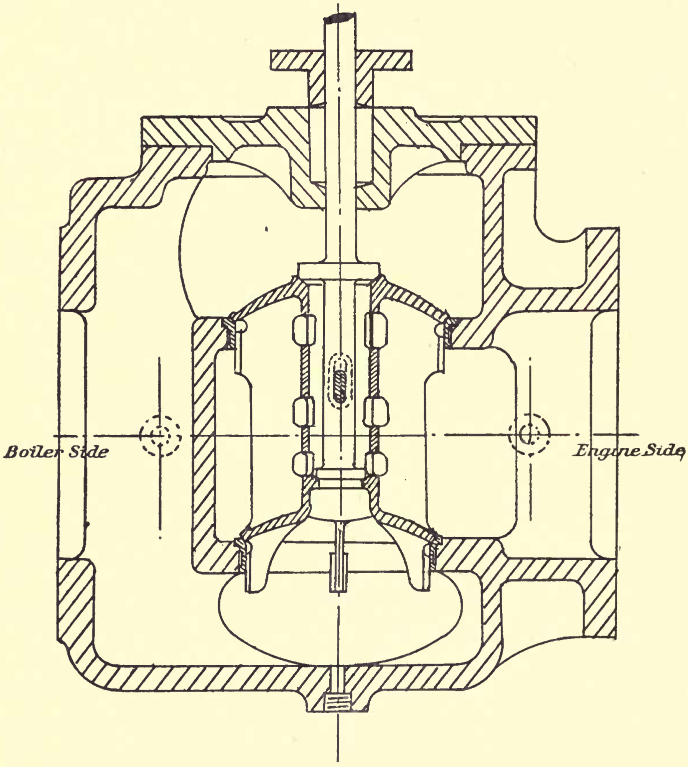


Fig. 6.

The first objection may be overcome by inserting a pilot-valve in the main valve, as shown by the accompanying figure (Fig. 5). On opening, the pilot-valve lifts about $\frac{1}{2}$ inch, before the main valve is moved, and a pressure on each side of the valve established. The second objection—that of a tortuous passage—can only be met by giving a large clearance past the valve. It will also be noted that the gland of this valve cannot be packed whilst under steam.

In winding engines and in others where a constant handling of the stop-valve occurs it is imperative that the valve shall be of the balanced type, and operated by a handle rather than a wheel and screw thread. The valve for such a purpose often takes the double beat or Cornish form, but is very liable to be leaky. The figure (Fig. 6) illustrates a form which was made for a large pair of winding engines, and is an instructive example of faulty design. The valve itself was of brass, sitting (or supposed to be sitting) on two brass faces forced into the valve

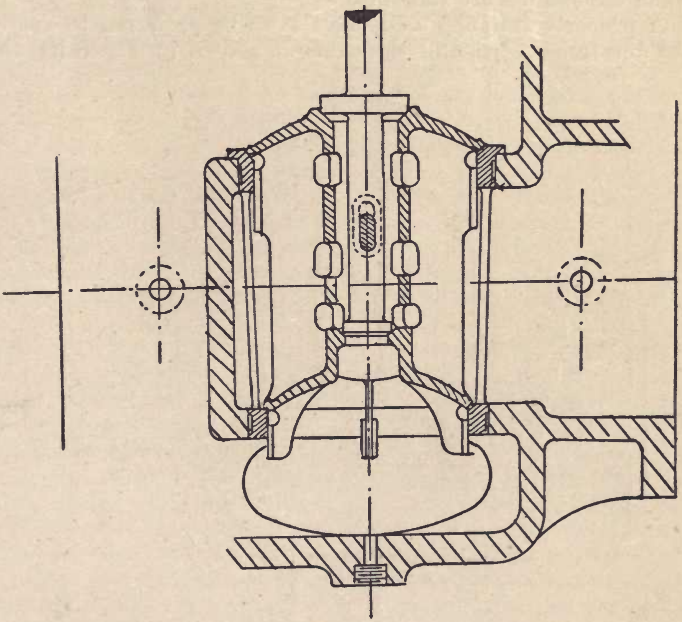


Fig. 7.

body. This valve could never be kept tight, and for good reason. When cold, the valve was carefully bedded so that each face was perfectly tight, but when under steam (the expansion of brass being greater than that of cast iron) the upper valve was horribly off, and enough steam at 120 lbs. came through to work a small engine. The seatings were then taken out and a new one fitted, as shown in the next figure (Fig. 7). This was a considerable improvement on the previous design, but not altogether satisfactory. The fact is that the distance

between the two valve seats was too great, and the least inequality of expansion of the seat and valve was sufficient to cause an escape. The crown of the top valve, which was not over stiff, and the under surface of which was exposed to pressure, may also have suffered a slight deflection from the great pressure, and as the lower valve was then hard on its seat it was impossible to screw it down any tighter. Most winding engines in the Lancashire coalfield district are fitted with this type of valve, but rarely is there a perfectly clear exhaust pipe when the engines are standing.

A properly designed slide-valve is to be preferred to one of the mushroom type, and the example shown in Fig. 8 has the

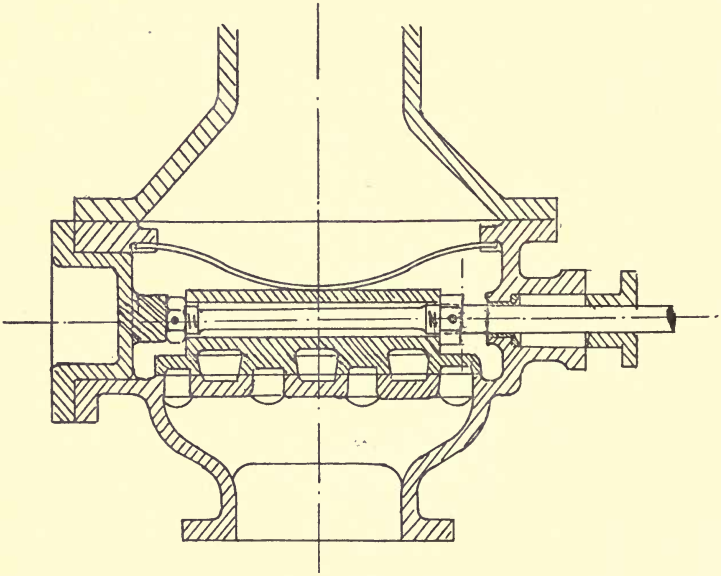


Fig. 8.

advantage of having a small unbalanced area. The gland may be packed under steam, and the whole arrangement is inexpensive.

Distributing Valves.—The distributing valves are the next moving parts of the steam engine after the stop-valve. This is not the place to deal with the subject of valve diagrams, but simply to discuss the various forms and defects of valves. The

reader is referred to works dealing specially with that subject in order to understand their action thoroughly. It will suffice if he has a general idea of the action of various types, and understands the terms lap, lead, compression, and so on. The aim of every designer of a new type of steam engine is, or ought to be, to place the cutting-off edges as near the cylinder as possible, or, in other words, to keep the clearance volume as small as possible. This is not so easy a thing as might be supposed; and every engineer finds—as most people do sooner or later—there is more wisdom required in the application of advice than in the giving of it.

Slide-Valves.—The slide-valve is the simplest and best valve considered merely by itself; but unfortunately it carries with it a large clearance and, under high pressures, it absorbs a large amount of power, or necessitates elaborate devices to render its action easy. It uses the same passage for the exit and entrance of steam, and owing to the shape of the ports it is difficult to scrape and smooth their surface, a defect which means a serious loss of efficiency. In horizontal engines its position is either at the side or the top of the cylinder, and therefore the latter is not self-draining. In spite of all this, however, it is more often used than any other type. Consisting of only one piece, and that of simple form, it is not to be expected that there is much scope for faulty design, so it may be passed over without further comment. As to its connection with the spindle, the use of nuts and a thread is the best method, provided the nuts are effectually checked against slacking back, because a slight adjustment of the valve is then possible.

Cornish Valves.—Cornish valves are much favoured on the Continent by such eminent firms as Sulzer of Winterthur; but in this country they are not much employed except by Messrs. Robey of Lincoln. It has been often stated as an objection that this type gives a large clearance, but if skilfully placed the clearance should not be so great as in a slide-valve arrangement, and if situated in the covers the clearance would compare favourably with a Corliss cylinder. The beats or faces are usually bevelled to 45 degrees, and vary in width from three-sixteenths to three-quarters of an inch. The narrower the face the better for tightness and balancing, but narrow edges are soon beaten down and require frequent renewal. A good form of drop valve might be designed if the beats were dispensed with, and the valve were allowed to drop inside the seating and remain suspended by an air buffer. It would then be in perfect equilibrium except for its weight; it would be quite noiseless in action, and if located in the covers the clearance would be

very small. A small spring would close the valve with great rapidity, thus enabling the trip gear to be made very light and compact. Some may object that it would be leaky, but accurate work in the machine shop would make the leakage quite inappreciable, or to take up the wear, a simple form of ring could be devised which would prevent any escape past the valve. A valve designed on these lines would compare favourably with the common form of Cornish valve.

With the advent of superheating it is probable that the double-beat valve will be more used than at present. Sliding pieces at the temperatures which will sometime be common will be perhaps the most troublesome feature in the new order of things; and as the old slides of gas engines were displaced by mushroom valves, so valves that have sliding action will be displaced by those of the Cornish type.

Corliss Valves.—The Corliss valve has many advantages over slide or piston valves, but most of those advantages are to be ascribed to the position it takes relative to the cylinder rather than to the valve itself. The usual disposition is to have the two exhaust valves at the bottom and a steam valve at each top corner, an arrangement which permits of efficient drainage of the cylinder without any special drain gear whatever. The steam valves at the top of the cylinder are constructed so that they will rise from their faces should the pressure in the cylinder exceed that in the valve chest, hence there is no need of relief valves. Further, the steam ports are used for steam only, and the exhaust ports only for exhaust. These passages are short and direct, and therefore clearance and condensing surface are small. Finally, the arrangement lends itself admirably to an automatic expansion apparatus and sensitive governing. The valves themselves call for no special comment, and the gear is dealt with in a subsequent chapter.

CHAPTER III.

CYLINDERS.

Corliss Cylinders.—The construction and design of Corliss cylinders vary considerably, but it is only the province of this work to notice those features that appear objectionable. The commonest objection is that of allowing the exhaust passage to pass along the barrel of the cylinder.

A better arrangement is to pass the exhaust through each foot and connect by a pipe to each end, as shown by Fig. 9. This is the plan adopted by Messrs. Hick Hargreaves. In this figure is also shown a cross-section through the valve-boxes, from which it will be seen that the clearance is considerable. In order to minimise this the valves may be placed in the covers and the clearance reduced very materially. This disposition of the valves has lately become very common, and in some cases projections have been provided on the pistons to fill the clearance passage. In doing this, however, some care must be taken to ensure that these projections allow sufficient passage for the steam to enter the cylinder at the beginning of the stroke, and it is also necessary to provide some means to ensure that the piston and rod shall have no circular motion on their axis, otherwise the projection will foul the covers.

It is a very common occurrence in engineering, as in other industries, that advantages are not always purchasable without sacrificing some other good feature, and the present is no exception, for with the valves situated in the covers the drainage of the cylinder is not effected by the exhaust valve, and special gear has to be provided. In vertical engines, however, this does not apply, and it is in this class of engine that the advantage of placing valves in the covers is most pronounced. Another objection to this arrangement is that the valve-gear has to be disconnected before the pistons can be overhauled.

There is one point about those Corliss cylinders which have cast ends at the front side which may lead to a serious mishap (see Fig. 10). The top half of the cylinder shows the piston and the end of the stroke, whilst in the lower section the piston

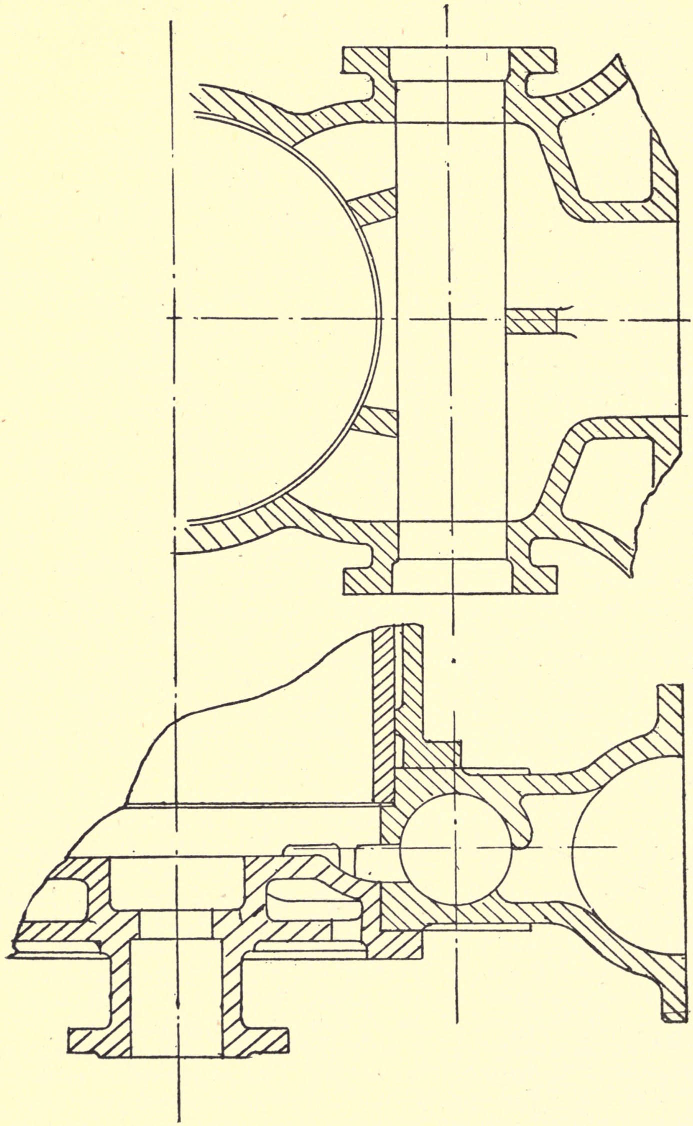


Fig. 9.

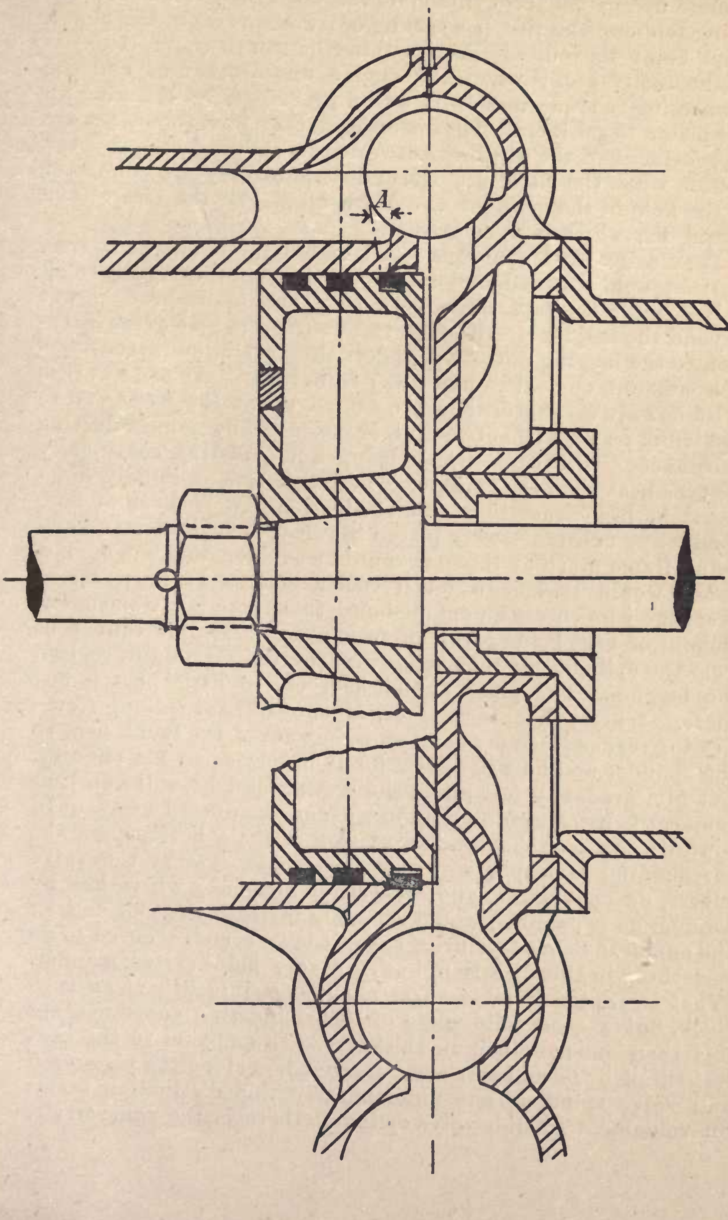


Fig. 10.

is close up against the cylinder end. Owing to the bellmouth being too long the ring has sprung out and prevents the piston from being moved. If such a thing happened in practice—it might easily occur when the engine was overhauled and the connecting-rod uncoupled—it would perhaps mean breaking up the piston to get it out. The distance, A, therefore, should always be greater than the distance between the cylinder end and the piston, when the latter is at the end of the stroke. The diameter of the hole in the cylinder end is regulated by the size of the boring bar which is to be used.

Insulation of Cylinders.—All cylinders should, of course, be thoroughly lagged, and all preparations for carrying relief valves, indicator taps, drain cocks, &c., should be brought out beyond the lagging. Although most engineers give great attention to the lagging of their cylinders they yet allow a considerable amount of heat to pass away from them. As most Corliss cylinders are arranged, they are bolted up at the front end to the trunk frame without the intervention of any non-conducting substance. The trunk frame being a big casting continually absorbs heat from the cylinder, where this heat is rapidly dissipated both by radiation and conduction. If a sheet of non-conducting substance were placed between the cylinder end and trunk frame much of this loss could be avoided, and the engine-house would become a much cooler place. It must strike everyone who enters an engine-house that there is a tremendous amount of heat being given off from somewhere. It cannot be from the cylinder for that being well covered feels quite cool on the lagging. The pipes are all clothed, so little heat is lost there. Where, then, is the great amount of heat coming from? Let the reader place his hand on some part of the trunk near to the cylinder and he will have a forcible answer to his enquiry. Let him gradually feel along the casting and he will find that the heat is being gradually radiated and conducted away until towards the crank shaft the casting is little hotter than the surrounding atmosphere. Sheeting placed against the valve covers, dashpot fixing, and, in fact, at all places where loss by conduction is possible, would certainly increase the efficiency of the engine. Some few firms have, it is true, endeavoured to do something in this direction, but in a very half-hearted manner. What is wanted is not a sheet about one-sixteenth of an inch thick, but a good solid piece of non-conducting substance not less than one-half inch in thickness. In addition to the heat lost through the cylinder ends, there is loss along the piston-rod and valve spindles, but unfortunately this dissipation seems unavoidable. In slide-valve cylinders there is the same waste,

because the feet conduct the heat to the bedplate and so all over the engine.

Connection of Cylinder to Bedplate.—At one time it was a very common practice to fasten the feet of cylinders to the bedplate by means of bolts and wedges (Fig. 11). This was bad practice, and was the cause of much trouble. The cylinder expanding more than the bedplate strained the feet enormously, and in some cases caused a fracture of the foot or the snugs on the bedplate, or else bent the cylinder slightly. Although this practice has now been abandoned, it is instructive to know of it, because there are many cases in which neglect of the consideration of the effect of expansion leads to disaster.

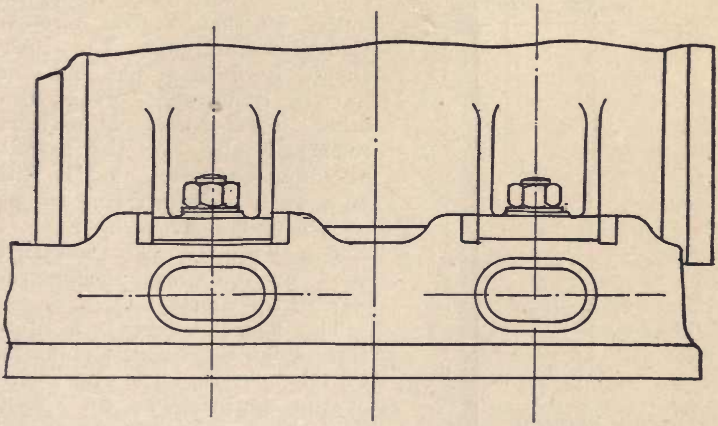


Fig. 11.

A very interesting case occurred some years ago near Rochdale. A large single compound tandem engine of the trunk design was put down to drive a cotton mill. The high- and low-pressure cylinders were mounted upon a massive cast-iron bedplate of box-section, secured to the foundation by many holding-down bolts. There was a small foot under the slides, but with this exception there was no other bearing part on the stone except the crank-shaft pedestal sole; the outer crank-neck was carried in a wall-box. This engine was truly lined up in erection, but, on starting, the crank-shaft necks heated tremendously, and it was some time before the cause was discovered. It was found that, as the engine warmed up, the crank-shaft, instead of remaining stationary, was moved slightly

so as to be no longer at right angles with the centre-line of the engine. The bedplate carrying the cylinder was too firmly secured to the foundation. Had this been a coupled engine, it is probable that no trouble would have occurred, but, as all the movement was at the end of the shaft, it was natural that the bearings should heat.

It is not the intention to discuss the advantages or otherwise of steam jackets, for that is a point on which the greatest authorities differ, but merely to state what should be kept in view should they be required. First, the jacket should be efficiently drained and the inlet should be well away from the outlet, so that a good circulation is maintained. The liner should be fixed at one end and have a little play left at the other for expansion. It may be remarked also, for the benefit of young engineers, that it will be a waste of time trying to get in position such a liner as is shown by Fig. 12. This was tried once, but no great success rewarded the efforts.

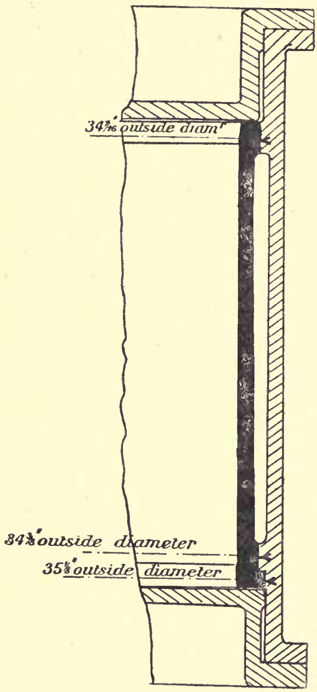


Fig. 12.

This figure also serves to illustrate a fault frequently found in cylinder covers. The upper part of the figure shows the spigot near the face of the flange, an arrangement that gives a considerable amount of additional surface to be alternately heated and cooled by the steam, besides increasing the clearance volume. The better construction is shown at the other end.

Bolts in Cylinder Covers.—In estimating the diameter and number of bolts required in a cylinder cover, the pressure of steam is not the only factor to be taken into account. An estimated stress of 5000 lbs. per square inch, reckoned at the bottom of the thread, is a fair allowance for bolts of 1 inch diameter, but this would be too much to allow on bolts $\frac{1}{2}$ inch or $\frac{5}{8}$ diameter, because the strain on the bolt due to tightening

up the nuts may be many times greater than that due to the steam pressure; indeed, it does not require a very strong pull at the end of a 12-inch spanner to break a $\frac{1}{2}$-inch bolt. Here a stress of 3000 lbs. per square inch would be the most that could be safely allowed. With bolts of larger sizes, the stress caused by screwing up does not bear the same ratio to the total strength of the bolt, consequently a greater pressure due to the steam may be allowed for, and a 2-inch bolt or stud may be stressed by the steam load up to 7000 lbs. per square inch without any risk. The pitch, again, has to be considered, or it may happen that, although the bolts are strong enough for the pressure, they are too far apart to prevent blowing at the joint. It is useless to give rules for the pitch of bolts, because the thickness of the flange and the quality of the metal have some influence. Experience alone can determine the most suitable pitch. The present object is merely to point out that there are other factors in the calculation besides the load on the bolts.

Relative Advantages of Studs and Bolts.—The question whether studs or bolts are the best for cylinder flanges is a nice one. Studs are a necessity in some parts of the flange, and many engineers employ them even where bolts could be employed. They give a narrower flange than a bolt, but are more costly to fit in, and, if broken when in position, cause much annoyance to remove. On the other hand, bolts are liable to drop inside the lagging when the covers are being removed, and in vertical engines, unless a ledge is provided, they may easily get pushed through, and the lagging will have to be stripped to recover them. The fillet at the root of the flange has also to be cuttered to give a flat surface for the head of the bolt. After all has been said, however, a bolt is a more mechanical fastening than a stud, and should be used wherever possible. The flanges of cylinders should be kept circular whenever possible, so that they may be machined when at the boring mill.

Large slide-valve cylinders usually have the valve-boxes cast separate from the cylinder, sometimes to diminish the risk of defective castings and sometimes to facilitate machining. In large sizes the distance from the face of the ports to the face of the steam-chest flange is too great to permit of the planing tool being steady when cutting.

Cylinder Details.—If lugs and projections have to be formed on the cylinder they should be kept back from the flange so as not to interfere with the path of the tool as it revolves round the edge of the flange. Fig. 13 will render this point clear. The tool clearance necessary varies, of course, with the form of cutter used, but in any case should not be less

than $\frac{1}{2}$ inch. In cramped situations the withdrawal of the valves and cylinders should be attended to, this consideration usually determining the nearness with which the cylinder can be brought to the wall or other adjacent parts. It is hardly necessary to say that all covers ought to have forcing-off screws,

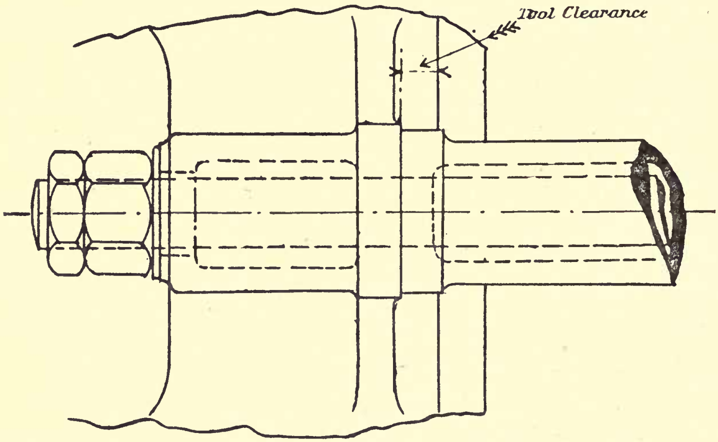


Fig. 13.

so that the joint can be broken without difficulty. The junk rings of pistons, delivery grids of pumps, and similar details should also be provided with forcing-off screws.

Relief Valves and Drains.—The drainage of cylinders is an important matter, the presence of water being both wasteful and dangerous, and a fruitful source of breakdown. Horizontal Corliss cylinders with exhaust valves at the bottom are self-draining, drain cocks and relief valves being therefore unnecessary. This is perhaps the most valuable feature of the Corliss cylinder. Other types, however, demand both drains and relief valves. The former should be coupled together, so that one handle, placed in a convenient position, operates both. A relief valve should also be provided on the steam chests of intermediate and low-pressure cylinders, whether Corliss or otherwise, and adjusted to lift at 4 or 5 lbs. above the intended working pressures of the respective cylinders, so that, in case of a late cut-off in the high-pressure cylinder, the pressures on the intermediate and low-pressure cylinders may not unduly stress the working parts affected.

Proportion of Cylinders for Compound Engines.—So many rules have been given for determining the proportion of the capacities of cylinders for compound engines that the writer hesitates to add yet another to the number; but, as the one used by him has proved so reliable and is not, so far as he is aware, generally known, it may perhaps be useful to some who are hesitating which method to adopt in their calculations. The method will be best illustrated by taking a practical example and working it through. Let it, therefore, be required to find the size of cylinders for a triple-expansion engine to develop 330 indicated horse-power, or thereabouts, with a boiler pressure of 160 lbs. per gauge, the revolutions to be 60 per minute, and the piston speed not to exceed 450 per minute. The engine will be of the Corliss type, with a high-pressure cylinder and the first low-pressure working tandem on one end of the crank shaft, and the intermediate and the second low-pressure working tandem on the other end; the cranks being at right angles.

It will be assumed that the engine is to have all the cylinders non-jacketed, but well lagged, with trip gear under the control of the governor for the high-pressure, and trip gear variable by hand in the case of the intermediate and low-pressure cylinders. The condenser will be the jet-condensing type, and the air-pump vertical and single-acting. It will also be assumed that the engine is a fair distance away from the boilers, say 150 feet, and the difference between the boiler pressure and the pressure in the valve-chest is 5 lbs. It will be seen later that it is necessary to take into account the type of engine and the general conditions stated, because these things modify the calculations.

Under the above conditions, the best terminal pressure in the low-pressure will be about 7 lbs. per square inch absolute. Theoretically it should be as low as the vacuum expected in the condenser, but as this is always an uncertain and variable amount it is best to be above it, otherwise there would be a danger of having a loop at the tail of the low-pressure diagram, and loops about there are very objectionable. The back-pressure in the low-pressure cylinder will be taken at 3 lbs. In most books on steam, a table of properties of saturated steam will be found, from which it will be seen that

The temperature of steam at 170 lbs. absolute = 368° F.
The temperature of steam at 3 lbs. absolute = 141° F.

The total range of temperature between which this engine is working is $368 - 141 = 227°$ F.; so that $\frac{227}{3} = 76°$ F. is the range for each cylinder. It is best to have the temperature ranges

equal in each cylinder, and, for the present, it will be assumed that there is no drop of pressure between the cylinders, the drop being accounted for later.

The initial temperature of the steam in the			H.P. cylinder			= 368.
The final	,,	,,	,,	H.P. ,,		$=368-76=292$.
The initial	,,	,,	,,	I.P. ,,		= 292.
The final	,,	,,	,,	I.P. ,,		$=292-76=216$.
The initial	,,	,,	,,	L.P. ,,		= 216.
The final	,,	,,	,,	L.P. ,,		$=216-76=140$.

Referring again to the table in order to substitute pressures for these temperatures,

The initial pressure in the	H.P.	cylinder	=	170 lbs.
The terminal ,,	H.P.	,,	=	60 ,,
The initial ,,	I.P.	,,	=	60 ,,
The terminal ,,	I.P.	,,	=	16 ,,
The initial ,,	L.P.	,,	=	16 ,,
The terminal ,,	L.P.	,,	=	7 ,,
The back ,,	L.P.	,,	=	3 ,,

From these results the ratios of expansion in each cylinder can be got, and are as follows :—

2·8 for the H.P. cylinder,
3·7 ,, I.P. ,,
2·3 ,, L.P. ,,

The expression $M = \dfrac{1 + \log R}{R} - B$ is sufficiently near for a calculation of this kind, where the exact value of several terms are not known, and a mathematically correct answer is not possible. The mean pressures for each cylinder are :—

63 lbs. H.P. cylinder.
22 ,, I.P. ,,
10 ,, L.P. ,,

The total horse-power required is 330, of which one-third should be developed in the H.P. cylinder, one-third in the I.P. cylinder, and the other third by the two low-pressure cylinders. For this the area of the H.P. cylinder should be

$$\frac{110 \times 33{,}000}{63 \times 450} = 128 \text{ square inches,}$$

and for the I.P. $$\frac{110 \times 33{,}000}{22 \times 450} = 366 \text{ square inches,}$$

and for the L.P. $$\frac{55 \times 33{,}000}{10 \times 450} = 403 \text{ square inches.}$$

Now comes the most important process in the problem, the

estimation of what will be the ratio of the actual indicator diagram to the theoretical figure on which the foregoing calculations are based. This opens a very instructive employment for the young student. For each class of engine this ratio will vary, and the only practical way is to compare the diagrams from actual engines working under known conditions. In the present example, which is an actual case, the ratios were—

·9	for the	H.P.	cylinder.
·87	,,	I.P.	,,
·88	,,	L.P.	,,

These ratios are known as diagram factors, and, as said, vary with the class of engine. Thus the factor is higher for a jacketed than a non-jacketed cylinder engine, higher for an engine with a large receiver capacity than one which has less, and so on. Unwin, in his treatise on machine design, has given a table of these diagram factors, but the student is strongly recommended to find his own; because he will then know exactly what conditions were present to give the results he arrived at.

Taking the above factors, then, for the case now under consideration, the area of high-pressure cylinders to allow for the losses due to withdrawing, compression, and loss of pressure between the cylinders must be—

$$\frac{128 \times 10}{9} = 142 \text{ square inches.}$$

Intermediate cylinder—

$$\frac{366 \times 10}{8 \cdot 7} = 420 \text{ square inches.}$$

Low-pressure cylinders—

$$\frac{403 \times 10}{8 \cdot 8} = 458 \text{ square inches.}$$

The diameter of the cylinders should therefore be $13\frac{1}{2}$ in. H.P., 23 in. I.P., and 24 in. each L.P.

This rule, it will be seen, gives approximately equal powers, temperature ranges, and piston loads for each cylinder. The only objection that can be urged against it—an objection which is likely to become stronger than at the present time—is that it is not applicable to engines using superheated steam.

CHAPTER IV.

AIR PUMPS AND CONDENSERS.

Types of Condensers.—The condenser and air pump are the next details to consider. Whether the condenser be surface or jet condensing depends upon the quality of the water, and the type and position of the air pump are determined by the level of the reservoir used for injection. Vertical air pumps are generally single-acting, horizontal air pumps both single and double-acting. Whatever the type, attention should be paid to the drainage question, and an uninterrupted fall from the cylinder to the condenser obtained wherever possible. With a horizontal condenser and air pump worked by a prolongation of the piston-rod it will not be possible to fulfil the above condition—a circumstance which in part accounts for the less perfect vacuum obtained in horizontal air pumps. All points in the pipes, from the cylinder to the condenser where air is likely to collect and lodge, should be carefully avoided, and where the water does not drain away drain cocks should be provided.

Jet Condensers.—Dealing first with the jet condenser the level of the surface of the reservoir from which the injection water is taken should not be more than 10 feet below the injection branch on the condenser. The bottom of the foot-box of vertical air pumps should slope towards the pump, so that no water lodges in the condenser; all air pockets ought to be avoided; and the clearance between the bucket and foot-valve should be kept as small as convenient. Wherever studs are used in the internal parts of air pumps they should be of gun-metal or delta metal with brass nuts, so as not to become worn and rusted together by corrosion. Brass liners in the barrel are advisable in cases where the water is bad, and the air pump-rod should be brass cased. The best type of bucket is one without packing, merely having grooves turned in the periphery; as a slight leakage of water past the bucket is not detrimental, and the fewer the parts inside the pump the better. A group of small valves in the bucket delivery and suction grids is better than one large one. In the first place, they are less sluggish in the action; and secondly, small sight holes can be

arranged for their convenient removal and renewal. Many condensers are faulty, in respect that there is no provision for convenient access to the valves. In some it is necessary to uncouple the lever, disconnect the vertical slides, remove the top cover, take out the delivery grids, and withdraw the bucket in order to renew the bucket valve. This means a great amount of labour, so that it is not surprising that many pumps are running with the bucket valves in shocking condition; and as a consequence the vacuum is exceedingly imperfect. The proper construction is to have a number of small valves in the

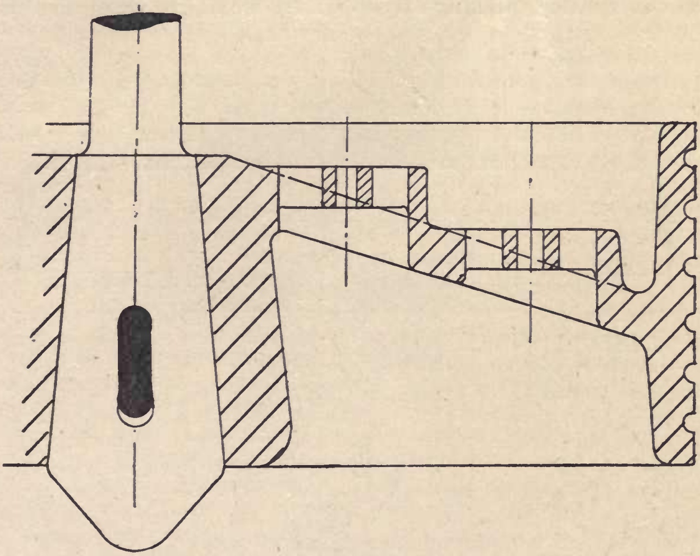


Fig. 14.

buckets, and to provide doors between the top of the barrel and underneath the delivery grid to enable them to be removed with small trouble. It is also good practice to construct the grids with sloping ribs, so that the water tends to rotate the rubber as it passes through the grid. The underside of the grids should be carefully rounded to break the shock where the bucket strikes the water. To this end the air pump-rod may also be shaped, as shown in Fig. 14, and the valves themselves may be arranged as shown.

It is remarkable what a small leakage of air into the condenser impairs the vacuum to the extent of 3 or 4 lbs. A

leaky stuffing-box alone is sufficient to render the condenser inefficient. In horizontal air pumps the gland and neck ring of the stuffing-box are soon worn down by the weight of the bucket and rod, and unless repeatedly packed and adjusted air will pass into the pump. It is easy to see when a gland is passing steam, but leakage of air into a pump is difficult to detect if not suspected.

One of the best methods of constructing the packing of an air pump-rod is to insert a "lantern" brass in the stuffing-box, and provide for a supply of water round the lantern. By this means a water-sealed packing is obtained, which is very efficient.

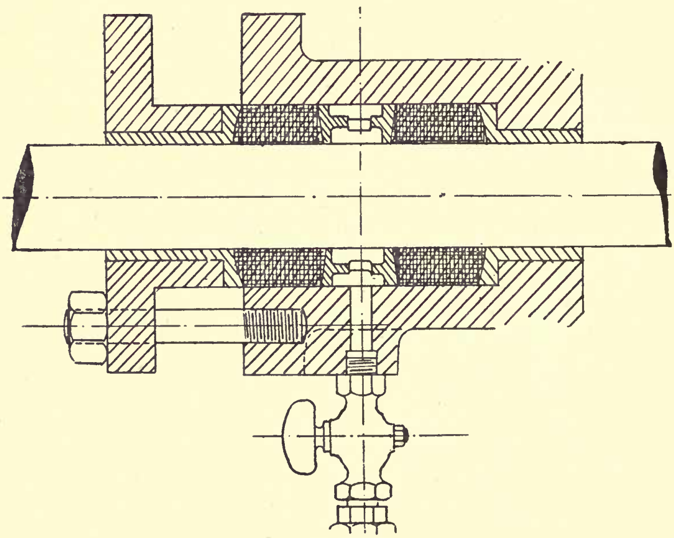


Fig. 15.

Fig. 15 shows the construction. The pipe is supplied from the hotwell, and the tap is useful when renewing the packing.

With respect to the velocity of water through the valves it will suffice if the speed be not more than 16 feet per second when the bucket is at its maximum velocity, and delivering its full capacity of water. This rule will determine to some extent the stroke of the pump, and, therefore, its diameter, for it will be found difficult to obtain more than about 30 per cent. of the total area of the bucket for water passage. Should the overflow in the hotwell be lower than the discharge for the water a cover will, of course, be necessary, but unless this condition holds, the

hotwell is best left open. The slides should be placed so that the delivery grid and bucket may be lifted clean out without disturbing any other parts. An important matter is the covering of the delivery valves with water, for unless covered there is some probability of air leaking past into the barrel of the air pump, whilst an excessive head of water on them causes a blow as the water strikes the underside of the valves. The best way to arrange this matter is to place the bottom of the overflow branch on the hotwell only just above the valve seats (say about $\frac{1}{2}$ inch), and make the opening long and narrow rather than circular; so that the water flows away as quickly as possible, thus relieving the delivery valves to the extent of the cover necessary for tightness. These remarks apply, of course, only to those air pumps in which there is a fall from the overflow to the reservoir. Many accidents have occurred through the nuts and cotters of air-pump buckets working loose, and being caught between the bucket and the bottom of the air pump. All loose pieces should therefore be carefully secured. Bolts or studs that are not disturbed in dismantling and overhauling should be riveted over the nuts; and those nuts which have to be unscrewed should be locked by copper pins. Cotters also should be secured in the same manner.

Surface Condensers.—In surface condensers the above remarks, for the most part, hold good. If the tubes are greater than 100 diameters in length they should be supported in the middle, otherwise they are liable to sag and crowd. From 2 to $2\frac{1}{2}$ square feet of cooling surface per indicated horse-power is the usual proportion, the larger proportion being used for hot climates. In spacing the tubes it should be seen that they are not pitched too closely, or the tube plates will be too weak to resist the outside pressure. The circulating pumps should, of course, be double-acting, so as to give a constant flow through the tubes. Small doors should be placed on the end covers so that any ferrule may be attended to without removing the end covers, and the position of the condenser itself ought to be such that the tubes can be withdrawn without interference with any part of the machinery. A manhole is generally placed for access to the body, and a preparation for a scouring cock should be provided. Where there are several engines working in close proximity to each other an independent condensing plant is advantageous. It will be cheaper than having a separate condenser for each engine, and will have the advantage that a good vacuum may be obtained at the commencement of working, by starting the condensing apparatus shortly before the main engines.

CHAPTER V.

MOTION WORK.

THE foregoing remarks dealt chiefly with the fixed parts of engines, or those parts which directly influence the course and action of the steam. The purely mechanical portions of engines now remain to be considered.

Piston-rods.—In designing piston-rods it is a good rule to allow a stress not exceeding 4500 lbs. through the weakest place, which is generally through the cotton hole, a rule which

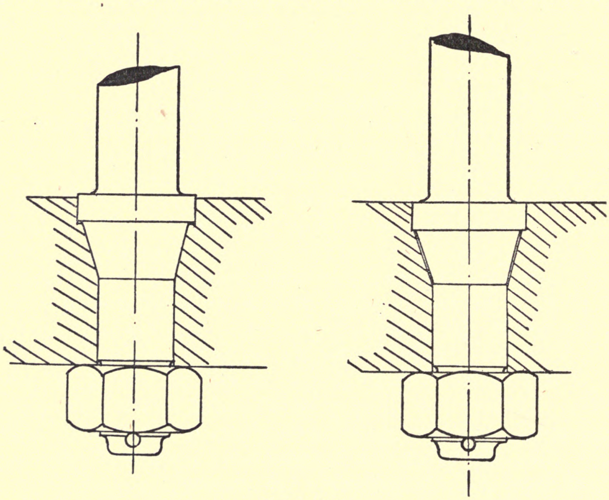


Fig. 16.

gives sufficient strength to allow for the length of the rod as well as the direct thrust of the pistons. The methods of securing the piston to the rods are numerous, some of them being not altogether satisfactory. A very common practice is to have both a collar and a taper, as shown by Fig. 16. With this arrangement it is difficult to get a good fit both on the collar

and taper; for should the hole in the piston be bored the least degree larger than the cone of the rod, all the bearing will come on the collar, and the taper part will be slack. On the other hand, when the hole in the piston is slightly less than the rod, the cone will be tight and the piston off the collar. The two conditions are shown exaggerated in the figure; the right-hand diagram being that in which the rod is bearing on the collar, and the left-hand figure when the cone is alone fitting.

Another common plan is not much better (Fig. 17). This design also necessitates both the hole and the rod being abso-

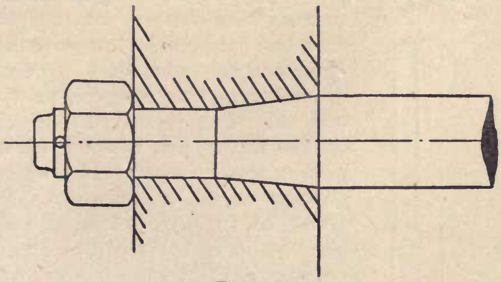


Fig. 17.

lutely correct, and when once in position there is no adjustment should the piston begin to knock and get slack on the rod. A better method is shown by the Fig. 18. This

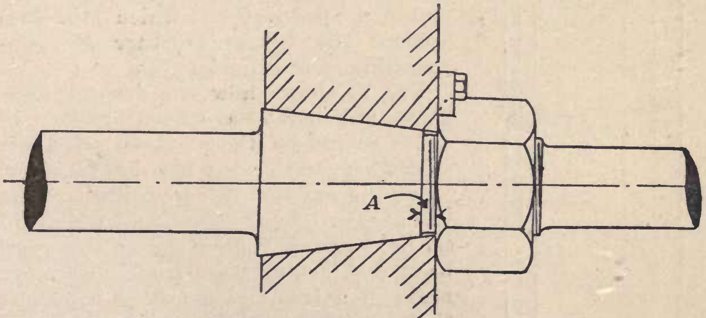


Fig. 18.

arrangement, in which the cone alone is employed, allows the piston to be forced up the taper until a tight fit is obtained, and should knocking occur, adjustment is possible, since the

clearance space, A, allows some movement on the cone. Where there is a tail-rod care must be taken that no part is too large to pass the bottom of the thread in the piston-nut.

In tandem engines great care is required in designing the piston-rods in order that they may be easily withdrawn, and that the covers, &c., will pass into their place. The accompanying diagram (Fig. 19), which represents the piston-rod of a large tandem engine, illustrates the matter very clearly. Starting at the back end, the load on the low-pressure cylinder was known, and that determined the diameter of the screwed portion of the rod, A. The rod then increased in size up the taper part, where the low-pressure piston was secured. B was made large enough to allow the nut to be brought along, or, in other words, the diameter at the bottom of the tread of the screw was slightly greater than the rod at C. Then comes another taper which determines the diameter of the rod at D. Thus it is seen that the size of the rod at the front end was not determined by the load in the pistons altogether, but by practical considerations which take into account the fitting together and dismantling of the engine.

Fig. 20 shows how the design of the above rod might have been modified in order to reduce the diameter, but this alteration carries with it a disadvantage. Here the rod is shouldered down to the size required by the load, and the gland bushes and neck rings of the high-pressure back and low-pressure front covers are made in halves and of such a thickness that their outside diameter exceeds the diameter of the shoulder on the rod. As before, the rod is passed through the high-pressure cylinder, and as soon as the end projects beyond the back of the

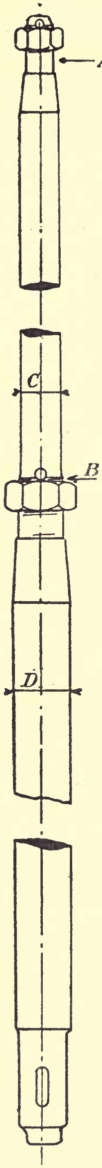


Fig. 19.

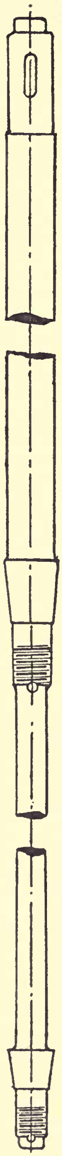


Fig. 20.

high-pressure cylinder the high-pressure piston and back cover and gland are threaded on. The rod is then pushed along until it passes through the low-pressure cylinder cover and gland, which are supposed to be in position, after which the low-pressure piston can be put in place. The gland bushes and neck rings being, as stated, in halves are then placed in their respective positions. It is an instructive exercise to consider cases of tandem engines in which the low-pressure cylinder is in front and the taper of the cone for each piston sloped in opposite directions. In such cases where the front of the low-pressure

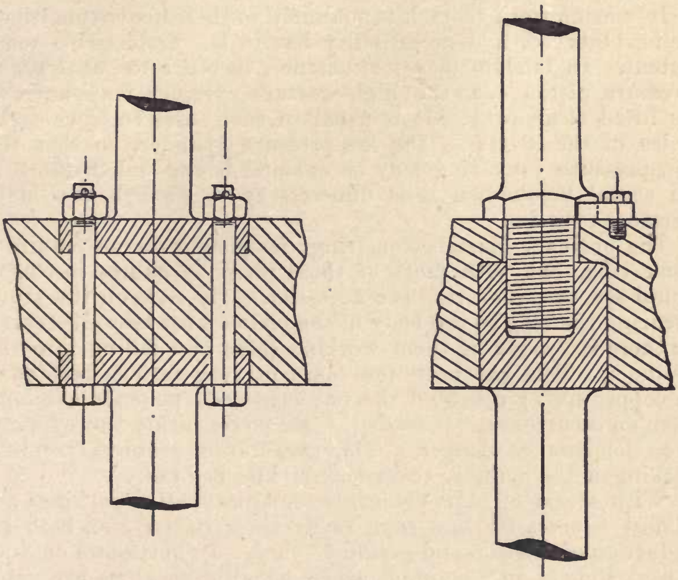


Fig. 21.

cylinder does not admit of the piston being put in from the front it will be found that putting the rod covers and pistons in their place is no easy matter, especially where space is limited. Placing a coupling between the cylinders simplifies erection but increases the length of the engine considerably.

If such a rod as that shown in the sketch is considered too long for convenient manufacture and handling, a joint may be provided at the front piston by either of the methods shown in Fig. 21. Then, if the cylinders have front covers cast with them, a common practice with those of the Corliss type,

the high-pressure rod may have to be put in from the back of the high-pressure cylinder, if the first method is adopted, and the low-pressure rod from the front of the low-pressure cylinder, and the engine must be arranged so that it is possible to do this. With regard to the distance between the two cylinders, it may be remarked that the ruling circumstance is the amount of room necessary to draw back the cover and piston-rod in order to give access to the cylinder and to make the joint for the cylinder cover. It will suffice if the distance between the cylinder flange and the piston, when pushed right back, is 15 inches or thereabouts.

In marine work room does not admit of these above conditions being observed, and accessibility has to be sacrificed to some extent. In tandem jobs, for instance, in order to get the low-pressure piston out, the high-pressure cylinder may have to be lifted clear away. It is usual in such cases to place sight holes in the covers of the low-pressure cylinder, so that the low-pressure junk ring may be examined, and unless there is an actual breakdown it is unnecessary to disturb the high-pressure cylinder.

The ordinary Ramsbottom rings for pistons do not require a junk ring, but with most of those piston packings having a spiral spring a junk ring is a necessity. The nuts on the studs securing the ring to the body of the piston must be well secured or there is danger of them working loose and falling into the cylinder. This has been the cause of very serious accidents. A copper split pin behind the nut is a good plan for avoiding such an occurrence. Indeed all loose pieces inside the cylinder are elements of danger. The piston-nut requires securely locking or the cylinder cover may be knocked out.

With steam of high pressures metallic floating packings are almost a necessity, and soon repay their initial cost, both by reduction of friction and saving of yarn. In horizontal engines a back slide is an advantage when metallic packings are used, because by careful construction the piston can be made to float in the cylinder, the rings alone pressing against the cylinder walls, the weight of the rod and piston being taken by the slides where ample surface and efficient lubrication can be obtained. Against this advantage has to be set the condensing action of a rod going in and out of the cylinder, so that it is generally found that back slides are only employed on the low-pressure cylinders, where they serve the double purpose of supporting the piston and rod, and for giving motion to the air pump.

Crossheads.—The types of crossheads are numerous, but may be broadly divided into two classes, those adapted for the old box

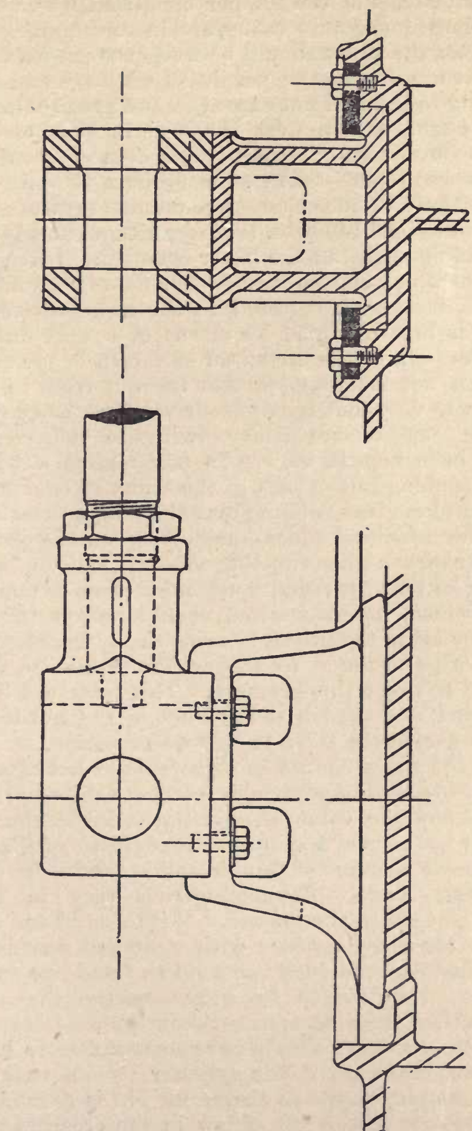


Fig 22.

form of bedplate, and the slipper crosshead which is used in marine engines almost universally, and in land engines commonly. The surface of the slides should be well grooved for oil, and the bearing pressure due to the weight of the rods and inclination of connecting-rod should not exceed 50 lbs. per square inch ; and the slides ought to run over the beds at each end so as to prevent the formation of ledges. The area of wearing surface on the crosshead pin—that is, the product of the length and diameter—should be such that the maximum pressure per square inch does not exceed 1400 lbs. for intermittent, and 1100 for continuous running engines ; the latter class being taken to include those engines which run for twenty-four hours without stoppage. When the crosshead is cottered to the rod, provision for disconnecting is best obtained by means of a thin nut and fine thread at the back of the crosshead as shown in the sketch. In all cases it is necessary to see that there is room to get at the end of the cotter, so that it may be driven back when disconnecting. Some engines are faulty owing to this consideration not having been kept in view. In trunk engines it is usual to provide an opening in the back of the trunk to enable this to be done. Even in engines running over there is at times an upward thrust on the crosshead slides—namely, when the piston lags—which often happens when shutting off steam, or when compression is excessive, so that provision must in all cases be made to resist this force, which, unless resisted, would be apt to strain the piston-rod or cause the piston to score the cylinder. In Fig. 22 the loose strips fastened to the engine frame by set screws are intended to resist this pressure. This crosshead is designed for a strap end, and the rib at the front of the cast-iron slide is cut away to enable the strap to be got in position.

Whether the slides should be adjustable or not is a matter of opinion. If the sliding surface is adequate the wear should be infinitesimal and the value of adjustment inconsiderable ; and unless under the care of a sympathetic engineer such adjustment is apt to become a source of trouble and excessive friction.

Connecting - Rods.—Connecting-rods vary in length as adjustment is made in the brasses. With a solid end the length increases as the brasses wear ; with strap and marine ends the reverse obtains. A rod with one solid end and one end strap is least influenced in length by wear, because the amount of alteration is then the difference between the adjustment of each end. This circumstance should have some influence in deciding the mechanical clearance in the cylinder. Solid ends of course can only be employed where the crank pin is overhung. The strap end requires less depth of jaw in the crosshead than the

marine type, whilst the brasses of the latter are easier to remove. When arranging the bedplate or other parts to clear the sweep of the rod, it is useful to remember that the worst position for some parts of the latter is not when the crank is at half stroke. A rod of rectangular section with the big end of the marine type is the best for high speed, the strap being liable to gape owing to the flexure of the rod at high speeds. The nuts of the cap bolts require careful looking after or they will shake loose.

Cranks.—The commonest shape for the overhung crank is a flat section. Yorkshire engine builders, however, largely adopt a turned crank like that shown in Fig. 23. This form

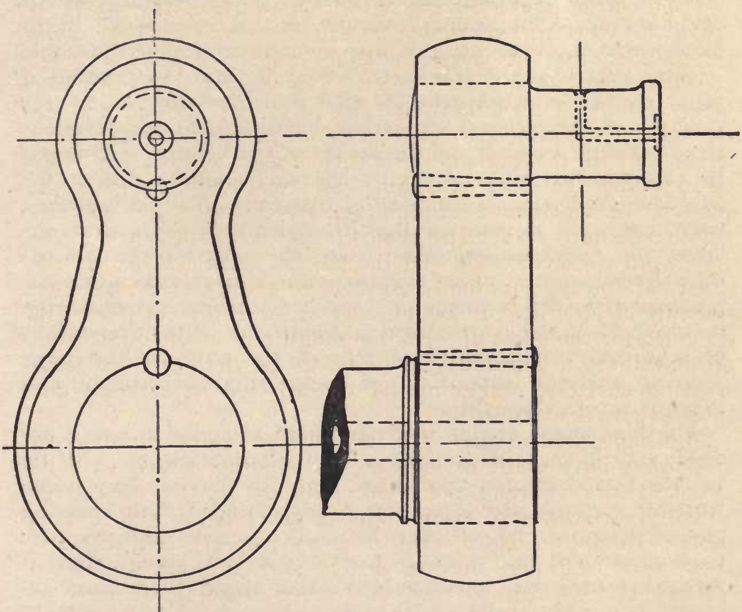


Fig. 23.

is neat in appearance, cheap to machine, and is altogether an improvement on the old form, and it is strange that it is not more extensively adopted. Mr. Michael Longridge, in a paper read before the Institute of Mechanical Engineers, which every engineer will do well to study closely, has shown that bad design in crank pins has been responsible for many serious breakdowns. The result of many observations of cases has shown that weakness is at the shoulder, when the pin is

smaller in the crank than at that part where the connecting-rod embraces it. The best practice is to make the pin larger in the eye of the crank and to provide a large fillet at the root, as shown in the figure. Mainly as a result of the adoption of this plan there have been no breakdowns due to faulty crank pins during the last twelve months in the engines under the inspection of Mr. Longridge, whereas in the previous seventeen years there have been no less than thirty-three failures.*

Of the methods of securing the pin to the crank shrinking and keying is the best, but the key ought to be of circular section. Some crank pins are secured by a cotter, an expensive method, and one which experience has shown to be unnecessary; others again are tapered in the crank, shrunk on, and rivetted over at the back, whilst others are secured with a nut at the back of the crank.

With regard to the crank pin, it may be said that like many other details of machinery its size does not depend on the strength of the material altogether, but on such considerations as coolness in working and durability of the brasses. As might be expected, the class of engine has also some bearing on the subject. A continuous-running high-speed engine, for instance, may have a pin of such size that the calculated pressure on the crank pin never exceeds 500 lbs. to the square inch. On the other hand, some winding engines have a calculated maximum pressure of 1100 lbs. on the pin, and are working satisfactorily. So much for the effect of working conditions on the dimensions of machinery. Durability is only to be obtained by large wearing surfaces, supposing, of course, that adjustment and lubrication are attended to.

The high-speed engine with its hundred hours' running per week may have a lift as long as the winding engine with its twenty hours' running per week only by having very large wearing surfaces and extra large proportions in the moving parts. Engines, indeed, are in many respects like mortals; they grow old, and become brittle and antiquated, and, if worked to the last, without any other cause than mere exhaustion of metal, collapse in some vital part. Use them well, let them have holidays, and they will live long; but place them in a dusty corner, and work them as only engines can work, and their life is soon spent. If you want a navvy of an engine you build it firm and colossal in its proportions, so that it may endure the heavy work it is designed for through a course of years; but if you have to make an engine for a light, active, and clean sort of job, you make it slimmer, and pay more attention to its appearance.

* *Vide* report of Mr. Longridge before the Steam Users' Association, 1900.

CHAPTER VI.

CRANK SHAFTS AND PEDESTALS.

Crank-Shaft Pedestals.—The main point to keep in view when designing a large crank-shaft pedestal is to see that the brasses can be removed with the least elevation of the shaft. To this end the bottom step should be circular on the outside, and steps to prevent rotation of the brass should be on one side only, so that when the crank shaft is jacked up, the step can be revolved until it comes to the top side, when it can be lifted clear away. Tapped holes, into which lifting eyes can be screwed, should be provided at suitable points both in the steps and caps. This question of lifting, by the way, is an important one, and applies to all parts of heavy machinery. Wherever there is a piece that either from its position or form does not allow of being slung, it ought to be provided with holes for lifting eyes; and such parts will include the large brasses, pedestal wedges, surface-condenser doors, and the pistons and rods of vertical engines.

The dead, unrelieved load of a crank shaft, on which is mounted a heavy wheel, requires not only ample bearing surface, but perfect alignment, if the neck is to run cool. Any sinking of the foundations is sure to cause trouble, unless the pedestal is made with a cradle, such as Messrs. Hick & Hargreaves, of Bolton, employ, whereby the steps are permitted to adjust themselves to the crank shaft. The bearing pressure of 160 lbs. per square inch due to the dead weight of the shaft, flywheel pulley, and cranks is good practice for heavy stationary engines. The bottom step should be well grooved to allow the lubricant to get beneath the shaft, and when soft white metal is employed the steps should be designed to give lateral support to the anti-friction metal on all sides.

On most recently-constructed engines of large size special arrangements are provided for the collection of oil from the crank-shaft necks, because it is found that the oil rots joints of the foundations, causing them to sink and crack the bedplates, or throw the engine out of line. This matter requires attention

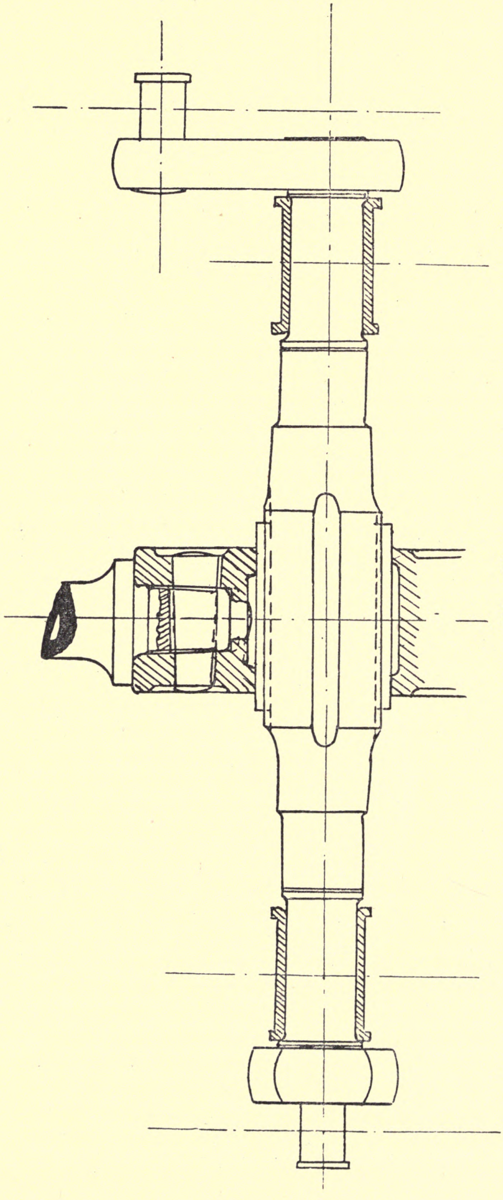


Fig. 24.

at other parts of the engine, and in Corliss engines it is usual to place dipper tins under the valve gear, as well as under the glands of the piston-rods.

Crank Shafts.—It is good practice to provide a small amount of side clearance or swim in crank shafts, so that any expansion due to heating of the bearings or other causes does not allow the collar of the shaft to bind upon the side of the steps. In single-crank engines some engineers make the crank-shaft neck without play, leaving the outer bearing free. In coupled engines a common practice is to make the outsides of each bearing without play, clearance being provided only on the inside, as shown in Fig. 24. This allows the shaft to expand without any fear of binding on the steps. It will suffice if this clearance be $\frac{3}{16}$ inch on each inside. At the points where the shaft is stepped up to a larger diameter, the fillets should have a good radius; certainly not less than $\frac{3}{4}$ inch. The object of the collar behind the crank in this and the previous figure is to give clearance for an oil catcher on the pedestal, whilst the small grooves on each side of the bearing prevent the oil running along the crank shaft on one side and the crank on the other, thus saving the foundations and other adjacent parts from being bespattered.

When the boss of the wheel is whole, as is common with large built-up wheels, it should be stated on the drawing of the crank shaft, so that there is no risk of both cranks being shrunk on before the boss is in position; and, in smaller pulleys, when the boss is in halves, secured together by means of circular hoops, these hoops have to be big enough to pass over the cranks. When it is inconvenient to do this, which might occur if the cranks are of the disc type, the hoops will have to be threaded on the shaft before the cranks are placed in position. In designing engines, the crank shaft is generally the first drawing made, since it determines the cross-centres of the engines, the position of the eccentrics, and the distance from the crank-pin to the journal. With broad rope pulleys the cross-centres of the engines are very much increased, the eccentric-rods or valve drive-shaft having to pass clear of the rim, sufficient clearance being provided to enable the ropes to be passed between the edge of the pulley and the eccentric-rod or shaft, as the case may be.

CHAPTER VII.

VALVE GEAR.

Eccentric-Rods.—Passing on to the valve gear, it may be remarked at the outset that more failures occur in this than in any other part of the steam engine, which is to be accounted for partly by reason of its complication, and partly by the severe and incalculable stresses to which it is subject. A common and unsightly defect in engines is vibrating eccentric-rods. These are often long and slender, and altogether too small for the work they have to do. It is difficult to get an eccentric-rod longer than 13 feet to work satisfactorily, unless made abnormally large. For anything over 10 feet it is an advantage to use hydraulic tubing for the barrel of the rods, welding on solid ends. For the same weight a much stiffer rod is obtained, though somewhat more expensive. Corliss engines above 3 feet 6 ins. stroke are generally provided with rocking levers, which serve not only to keep the eccentric-rods of reasonable length, but to reduce the overhang of the valve gear. These rocking levers need to be very stiff to withstand the severe twist, which is reversed every stroke. The alternative to a rocking lever is a cross-shaft driven by bevel wheels from the main crank-shaft. When carefully planned, the cross-centres of the engine may be less than an eccentric and rocking lever drive in cases where the rope pulley is wide; but, on the other hand, it is more noisy, and, unless the teeth of the wheels are machine-cut, there is some irregularity in the drive due to the loss of motion in the wheels and the twist of the shafts.

Linkwork.—The linkwork of all well-designed engines is arranged so that it can be dismantled with little difficulty. When overhung pins are employed, the rod should be made to draw off in the same direction at each end, as shown by Fig. 25. Fig. 26 shows the faulty arrangement which necessitates uncoupling each rod end in addition to taking off the collars.

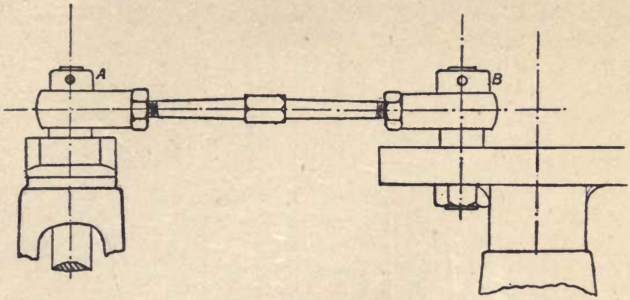


Fig. 25.

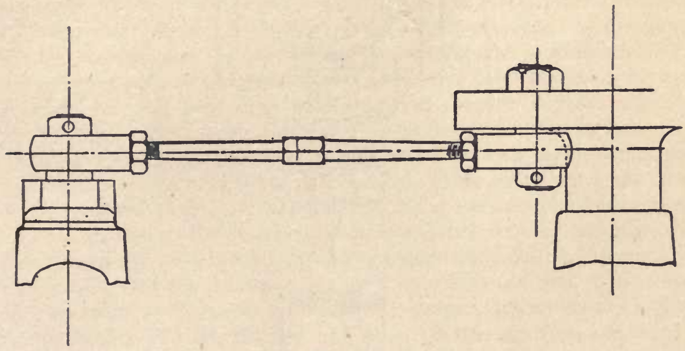


Fig. 26.

Rod Ends.—The designs of eccentric- and valve-rod ends are various. Of the types illustrated in Fig. 27, that shown by A is expensive, and the cotter prohibits its use in some cases in Corliss gear. It has this advantage, however, that, providing the collar of the pin will pass through the slot in the rod, that collar may be one with the pin, since the cotter and block keep the rod in place. The next type (Fig. 27) is cheaper and neat, but necessitates a loose collar on the pin, big enough to cover the brass, otherwise there is nothing but the friction between the brasses and the rod to prevent the latter sliding off the pin. After adjustment has been taken up, the whole thrust of the rod comes on the set screw, which, in consequence, is apt to become worn and stripped. The marine end is a good type, but expensive. Of the three designs mentioned, the first is, on the whole, most convenient, and is the one most often used.

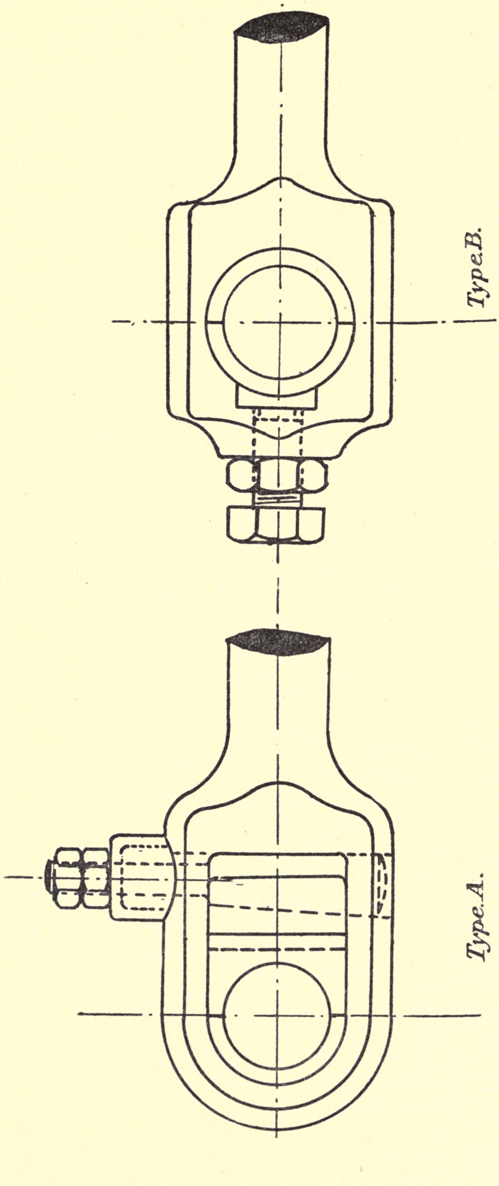


Fig. 27.

Movement of Parts.—An important point to be kept in view in all classes of valve gear is to reduce the motion of the parts as much as possible. This is to be obtained by double and multiple ported valves working across long and narrow ports. In link motions it is specially advantageous to reduce the motion of the parts, because, unless the length of the link is from $2\frac{1}{2}$ to 3 times the valve travel, there is considerable slip in the link block, and the angles become unsatisfactory; and, on the other hand, a long link is both expensive and inconvenient.

Force Required to Work Valves.—All pins and eyes for link motion should be case-hardened since adjustable ends are not convenient. The area of the link block may be such that the calculated bearing pressure on the pins is not greater than 160 lbs. It is impossible to estimate correctly what will be the force required to actuate a slide valve, so that, to allow for the worst case, it is usual to assume a coefficient of friction of 0·2, and calculate the whole area of valve face as unbalanced, irrespective of the fact that the steam ports, to a small extent, relieve the valve. In a series of elaborate experiments, Mr. Aspinall, of the Lancashire and Yorkshire Railway, found the coefficient of friction of a locomotive slide-valve very small, so small indeed as to be represented by a decimal in the second place; but neither Mr. Aspinall himself, nor any other engineer has yet adopted that coefficient as a basis for calculation. Valve gear is not like a bridge, for instance, or a boiler, where the strains can be worked with some degree of accuracy. The designer has to trust to his eye and his observation of similar cases to arrive at the dimensions of various parts.

In those valve spindles which do not run through the steam chest at the back end, there is an additional force thrown on the rods and pins, due to the unbalanced area of the valve spindle. Thus, in the case of an engine working with 150 lbs. steam pressure and having a valve spindle 2 inches diameter, there is a force exceeding 4 cwts. resisting the motion of the valve in one direction, due solely to the end pressure on the valve spindle. This force alone is considerable when it is remembered that it has to be overcome by the *thrust* of long rods and spindles, and ought to be considered when designing the motion work.

One chief draughtsman of the writer's acquaintance had a very peculiar method of designing valve gear. He would have the gear set down to please his eye, which, it must be admitted, was a well-trained one in some respects, and then he would proceed to calculation, and, strange to say, whatever results the calculations gave he always maintained that it was satisfactory, and that the gear was in accordance both with the figures and his

sense of proportion. In a word, his method was to fit a rule to the case and not a case to the rule. This method of procedure is too common in engineering, and is not mentioned with the intention of deprecating theory, but to show that theory does not go far enough, and begets distrust in the minds of practical men. Take the piston valve, for instance. Beyond the stress produced by the momentum of the valve itself and the friction of the rings, there is apparently no other force causing resistance to its motion, yet in practice the gear for working the piston valve, at least in marine work, is not visibly less massive than if the valve were of the ordinary type. Why is this? The answer is to be sought in the fact that a conviction has taken hold of the minds of practical men that there is some force not accounted for by theorists, some force that may not always be present—nay, which rarely is present—but which has to be met some time or other. What the force is they would not be able to say. Ignore it and they would prophesy breakdown, and their fears would not usually prove groundless.

Corliss Valve Motions.—In Corliss gear, theory is very little requisitioned. Each maker adopts his own proportions and designs, arrived at in many cases by accident and oftenest by appearance. There are, however, some general lines to be observed to ensure satisfactory working.

In the skeleton diagram of the wrist-plate motion in Fig. 28 the extreme angles of the motion are shown by dotted lines, the central positions by heavy lines. The angles, A, should not be less than 45 degrees, nor the angles, B, more than 135 degrees, or the rods may get too straight to work easily, and any wear in the pin at C would make matters worse. The design at the rod end will determine the shortest distance between the wrist-plate pins, D. Their position, with respect to the horizontal centre line, should be such that when the extreme position is attained the wrist-plate rod does not cross the line joining the centre of the wrist-plate and the pin of the valve lever; otherwise chattering is liable to be set up in the wrist-plate pins, due to the frequent stress reversals.

When the pin of the valve lever comes under the bridle, as in Fig. 29, there is another matter which may limit the motion of the valve lever. The sketch is self explanatory, and it only remains to say that the designer should make sure when laying down the wrist-plate diagram that the details of the bridles and rod ends will enable the motion to be obtained, or he may find that when he has half finished the gear he will have to begin again with longer valve levers and increased movements.

Wrist-plates are neither necessary nor desirable adjuncts to a Corliss trip gear. They increase the number of parts and consequently wear and tear. The advantage claimed for them—that

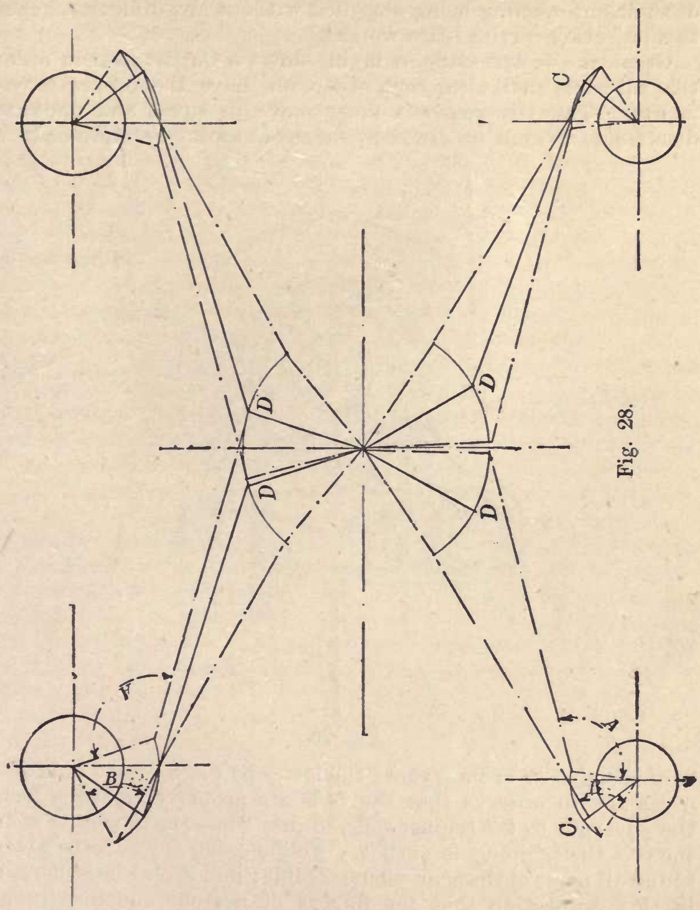


Fig. 28.

they give a better valve movement—scarcely compensates for the objections. In a few cases, notably with the Inglis & Spence gear, a wrist-plate is necessary to give support to one end of the slip-rod, but the majority of gears can be arranged without one.

It may be further urged that the dwelling angle, and consequently the total movement of the valve may be reduced by the use of a wrist-plate. Double ported valves, however, permit of sufficient opening being obtained without any difficulty, hence this objection carries little weight.

Great care is necessary in laying down a Corliss gear in order that all parts shall clear each other and have the proper movements. The tripping-rods must move in equal and opposite directions, a result obtained by means of toothed sectors or by a

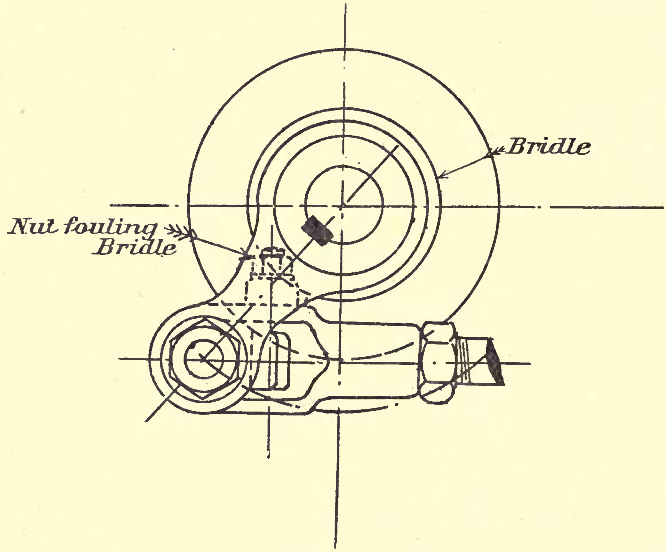


Fig. 29.

double-ended or beam levers; the latter by preference. It is also necessary to observe that the rods are properly arranged from the governor to the tripper-rods, so that when the governor is on the rise the tripping is earlier. Finger room ought to be given round all parts of the gear when possible; and it may be remarked in this connection that the fingers of a stout and oleaginous engineer are generally much fatter than those of the designer.

The aims to be kept in view in bringing out a good Corliss gear are, firstly, to have the piece carrying the catch as small as possible. This will ensure rapid engagement and higher speeds, because the inertia of a small piece is more easily controlled by the engagement spring than that of a heavy one. Secondly, to

arrange the engagement pieces to remain suspended when disengaged, and not to strike upon a pad or metallic surface. By these means the gear would be practically noiseless in its action, and the wear slight. Finally, the rods should be in tension when pulling against the dashpot springs. The writer is not aware of any gear in which all these considerations have been regarded, and suggests them as a profitable field for study and invention.

Valve Levers, &c.—The use of turned valve levers instead of those of flat section is advisable, both on account of appearance and cheapness of manufacture. They are also easier to keep bright. Bridles without stuffing-boxes are becoming common, and when carefully designed and constructed are quite satisfactory. They also carry this advantage that a much shorter bridle can be obtained than when a stuffing-box is used; for where a rocking lever is introduced it is generally the bridle that determines the overhang of the gear, and not the position of the eccentrics on the crank shaft. The end of each steam valve spindle should be provided with a square in order to bring the catches into engagement at starting by means of a key fitting on the square. Sometimes the dashpot lever has a projection formed on it for this purpose. Some may think all these are trivial matters. In truth they are; but it is the minor details of machinery that occupy most of the time and care of engineers, and it is chiefly in small details and improvement in methods of construction that further advance is to be sought. In all essential parts the steam engine of to-day is as Watt left it, yet if that famous man could revisit "this dim spot which men call earth," it is probable that he would find himself behind the times, and inferior to the present-day engineer, in details only, and not in the general principles of engine design.

CHAPTER VIII.

LUBRICATION.

Cylinder Lubricators.—The practice of engineers in the lubrication of cylinders varies widely. Marine engineers rely mainly upon the wetness of the steam for lubrication, and seeing that the pistons are vertical, and that, except in a heavy sea, there is no side pressure on the walls of the cylinder other than the spring of the rings, it is not surprising that internal lubrication is dispensed with, especially when it is remembered that the condensed steam is continually returned to the boilers. Few stationary engine-builders, however, would care to dispense with a sight-feed lubricator on high-pressure cylinders.

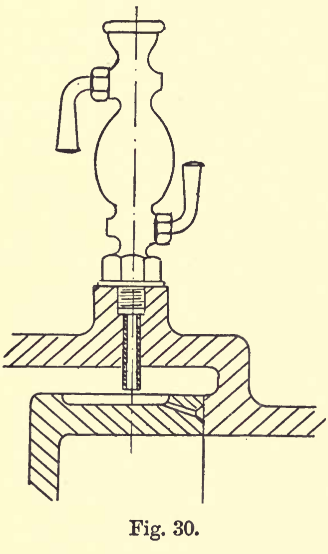


Fig. 30.

Of sight-feed lubricators, those having a positive action are the best. The ordinary displacement type is apt to choke up. The feed should be before the stop valve, so that all the working parts in connection with the cylinder receive the benefit of the lubrication. In addition to the sight feed, tallow cups should be placed so that a flush of oil can be poured on the valve faces in case of the valves groaning, an occurrence which sometimes happens when engines are started after having stood for a few days. The holes of most tallow cups and other lubricators are too small, and soon become clogged up. None should be less than three-sixteenths of an inch diameter. The oil from the cups should be led by a short piece of pipe, or by a nozzle on the shank, close to the desired point. If the shank simply

screws into the casting, the oil may cling to the sides of the chamber and run where not required. Fig. 30 shows how the shank of the cup should be arranged.

Crosshead Lubricator.—The crosshead pin is generally lubricated by means of a syphon cup, which is sometimes placed on the connecting-rod, and sometimes on the crosshead, as may be convenient. Another plan which answers well is the wiper arrangement, in which a lip on the oil cup on the crosshead pin licks an oiled wick suspended from another cup placed near the end of the crosshead's limit of motion. The crosshead slides may be fitted with brass combs, which dip into oil wells at the end of the slides, and spread the oil over the surface of the slides on the receding stroke. Grooves are also cut in the slides to facilitate the oil spreading over the whole surface.

The brasses of the crosshead pin are sometimes cut away at the part which takes the least thrust, thus forming a space for oil, and sometimes a flat is formed on the pin itself for the same purpose. From these recesses grooves are cut round the brasses to allow the lubricant to reach that part of the journal which takes the greatest thrust. These grooves should either be blind or return to the recess, so that the oil cannot escape except by passing between the brass and the pin, and the edges should be well rounded to avoid scoring the pin.

Lubrication of the Crank-Pin.—The efficient lubrication of the crank-pin is of great importance. The centrifugal lubricator is usually employed, and appears to be very satisfactory. In overhung cranks it consists of a small brass saucer revolving on the centre of the shaft, and connected to the crank-pin by a tube. The oil is fed to the saucer from a cup fastened on a hand-rail pillar. The oil supply is visible. The oil passes up the tube by reason of centrifugal force imparted by the revolution of the tubular arm, and is led to the pin through holes drilled therein. When the crank is of the double-sweep type, the saucer is annular in shape, and is placed behind the web of the crank.

In addition to the centrifugal lubricator an oil cup is often placed on the connecting-rod as a supplementary feed. In vertical engines it is placed high up on the rod, and a copper pipe leads to the crank-pin.

The remarks on the oil-ways for the crosshead-pin apply equally to the crank-pin. When white metal is employed the brass must be formed to give support to the metal on all sides, and the soft metal alone should touch the pin. A slight end-play is advisable, so that the connecting-rod may adjust itself to the variations of the length of the crank-shaft. When oil-ways are

formed in white metal they should be wide and well rounded on the top, or the spreading out of the metal may close them (see Fig. 31).

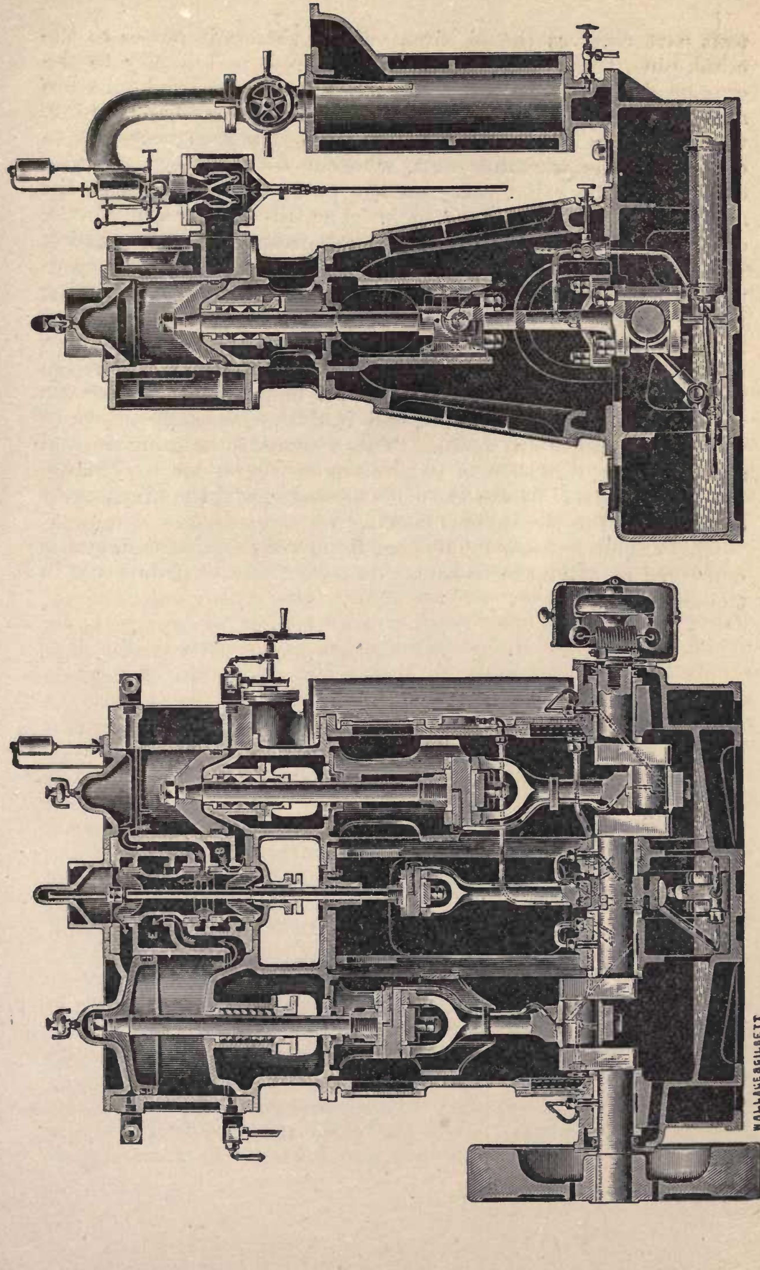
Fig. 31.

Crank-shaft Lubricator.—The lubricating arrangements for a large crank-shaft journal are sometimes elaborate. A small pump driven by a cord on the shaft draws oil from a tank at the base of the pedestal, and delivers it to an oil-box on top of the pedestal. The oil is then fed to the bearing through taps, so that the drip is visible. Should the supply from the pump exceed the amount passed by the taps, the surplus is returned to the tank by an overflow pipe. After the oil has passed through the bearing it is collected by drippers at each side of the journal and returned to the tank, where, after running through a sieve, it is ready to be pumped to the oil-box, thus being used over and over again. This arrangement, although costly, is very satisfactory, and soon repays the initial expense by the saving of oil effected.

In small high-speed vertical engines the lubricating arrangements are not so elaborate. A large oil-box is usually fixed to the side of the cylinders, from which taps with a visible drip supply oil to pipes leading to the various bearings. The pipes should not be less than three-eighths of an inch bore, otherwise they are liable to become choked up very frequently. The drip from the bearings is collected in the crank pits, and before being used again should be well strained and separated from the water which escapes from the glands.

With the exception of the eccentrics and the eccentric-rod ends, it is not usual to provide lubricators for the details of the valve and governor gear. The rod ends of these details, however, should always be provided with oil holes, well countersunk at both ends. The move of these parts being small they can usually be lubricated whilst in motion, and as the work on them is small they require little attention. The wear is more important, and may cause some irregularity in the running of the engine. The surface area should therefore be liberal.

Forced Lubrication.—The lubrication of enclosed dust-proof high-speed engines differs from that of an ordinary engine in that the oil is supplied to the bearings under pressure. A small pump worked from the crank-shaft by an eccentric supplies the pressure. The crank-shaft being the most impor-



(a.) (b.)

Fig. 32.

tant part receives the oil first. From thence it passes to the crank-pins, then along the connecting-rod, and finally to the crosshead slides. In some designs the eccentric-rod pins are also lubricated under pressure. When the engine is running, the oil squirts from all the bearings, so that it is necessary to enclose all the working parts, whereby dirt and grit are also excluded. The relief valve on the pump is usually set to lift at about 40 lbs. per square inch. The oil-ways are, of course, designed so that the lubricant cannot escape except by passing between the shaft and the bearing. This system requires accurate workmanship in the wearing surfaces and close-fitting brasses, or the oil escapes too freely, and some of the bearings fail to receive a proper supply. Careful attendance and adjustments are necessary; and in order that these may not be too frequent, large bearing surfaces should be provided. After use the oil falls to the crank pit, and is drawn through a sieve to the pump and used again. The general arrangements and appearance of this class of engine are clearly shown by Fig. 32. The pump is driven from the main eccentric, and the oil passages can be traced to the various bearings.

This engine is made by Messrs. Belliss of Birmingham, and is employed to drive the dynamos on one of the torpedo boats in the British navy.

CHAPTER IX.

MISCELLANEOUS DETAILS.

Holding-down Bolts.—In arranging the holding-down bolts for an engine it is well to see that they can be dropped in position after the engine is erected, and if this is not convenient or possible the erector should be informed, so that he may place the bolts in the foundations before setting down the bedplates, otherwise there is danger of very bitter things being spoken and thought. When employing rag bolts it should be remembered that they can only be used where the piece to be secured can be dropped straight down over the bolts, because they must be leaded into the stone before the piece is in position. For similar reasons studs are not admissible in some places, and set screws have to be employed.

Handing of Parts.—As it is well known by all who have practical acquaintance with machinery some pieces are suitable for either the right or left hand, whilst others are one hand only.

It requires some experience to determine rapidly whether a certain portion will hand about, and want of this experience is a frequent cause of mistake. A slide-valve cylinder with feet cast on will, of course, not hand about, because of the feet and the steam and exhaust branches. A Corliss cylinder that has no preparation for a wrist-plate will hand, with the exception, perhaps, of the indicator bosses and the preparation for carrying the dashpot. It is such slight differences as these that are apt to be overlooked, and it is a good plan to have the small preparations on each side, so that the pattern serves for either hand without alteration, even to such minor details as indicator bosses. The best way to determine whether a piece will hand is to imagine it cut in halves down the centre. If there is any difference whatever in the halves then the piece will not hand. In pipe flanges a somewhat similar occurrence takes place when the number of holes is 6, 10, 14, 18, and so on, progressing by fours. With two bends at right angles the holes in one flange will be off the joint of the pipe or templet way, whilst in the other flange abutting thereto two holes will lie in the joint of the pipe. But when the number of holes is four, or any number divisible by four without a remainder, the bends will turn at right angles to each other without any difference in the position of the holes in each flange with respect to the joint of the pipe. It is easy to multiply instances of this nature, and a

profuse writer could make many chapters on the subject. The point to be impressed, however, is the necessity of unceasing vigilance. Suspicion, though not an amiable quality in general life, is a professional attribute of great value; and this attitude towards all new devices and contrivances—the conviction that if there is the least loophole they will cunningly evade their duty—is the only way to eliminate failures, breakdowns, and imperfect designs; and the story of the man who built a big drum in a room with a small door and window is instructive, and should ever be in the minds of young engineers.

Conclusion.—Having thus dealt with the steam engine in a practical manner, it may not be out of place to sum up the points which ought to be kept in view in seeking increased efficiency.

The great cause of waste or condensation in the steam engine is liquefaction in the cylinder, and its prevention is one of the most difficult and fascinating problems that the engineer has before him. In seeking to reduce this liquefaction, superheating steam jacketing, heat insulation, efficient drainage, smooth surfaces in the ports, as well as on the pistons and covers, must all be carefully thought out. In present-day engines the clearance given for mechanical reasons between the pistons and covers is nearly always more than need be if the pistons and covers were trued up on the face, and the connecting-rod were designed so that the wear on one end counteracted the wear on the other, thus maintaining a constant length between the centres. When this is done it will be important to take into account the expansion of the piston-rod due to the heat of the steam, more especially in long-stroke tandem engines, more clearance being given at the back end of the cylinder, so that when steam is turned on the expansion will equalise the clearness.

But of all the directions in which increased economy is to be sought, superheating appears to offer the greatest promise, especially seeing that all forms of soft packing can be dispensed with, and oils obtained which are efficient at a temperature of 800° F. With superheating the importance of improvement in heat insulation and clearance will become increased.

It is not to be expected that much advance in the direction of reduced engine friction will be made, the saving by employing ball-bearings scarcely being justified by the present price of coal and the complication and expense incurred.

THE END.

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